Refrigeration

Professional Engineer Examination Review – Mechanical;

Center for Continuing Engineering Education (C2E2)
web.odu.edu/engr/ccee

KEY

P.E. REFRESHER - REFRIGERATION

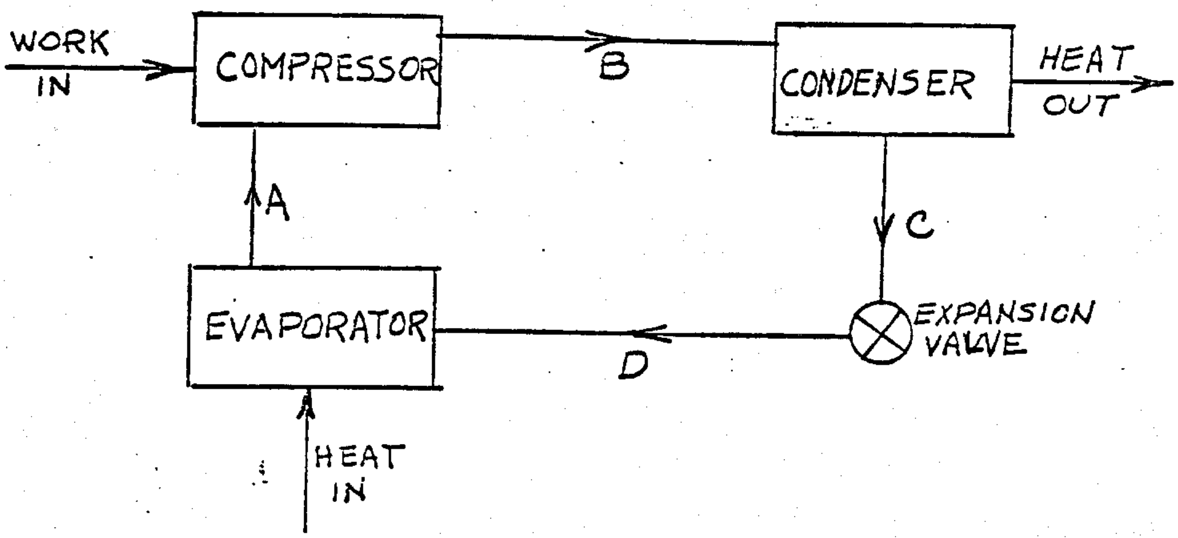


Fig. 1 Refrigeration System

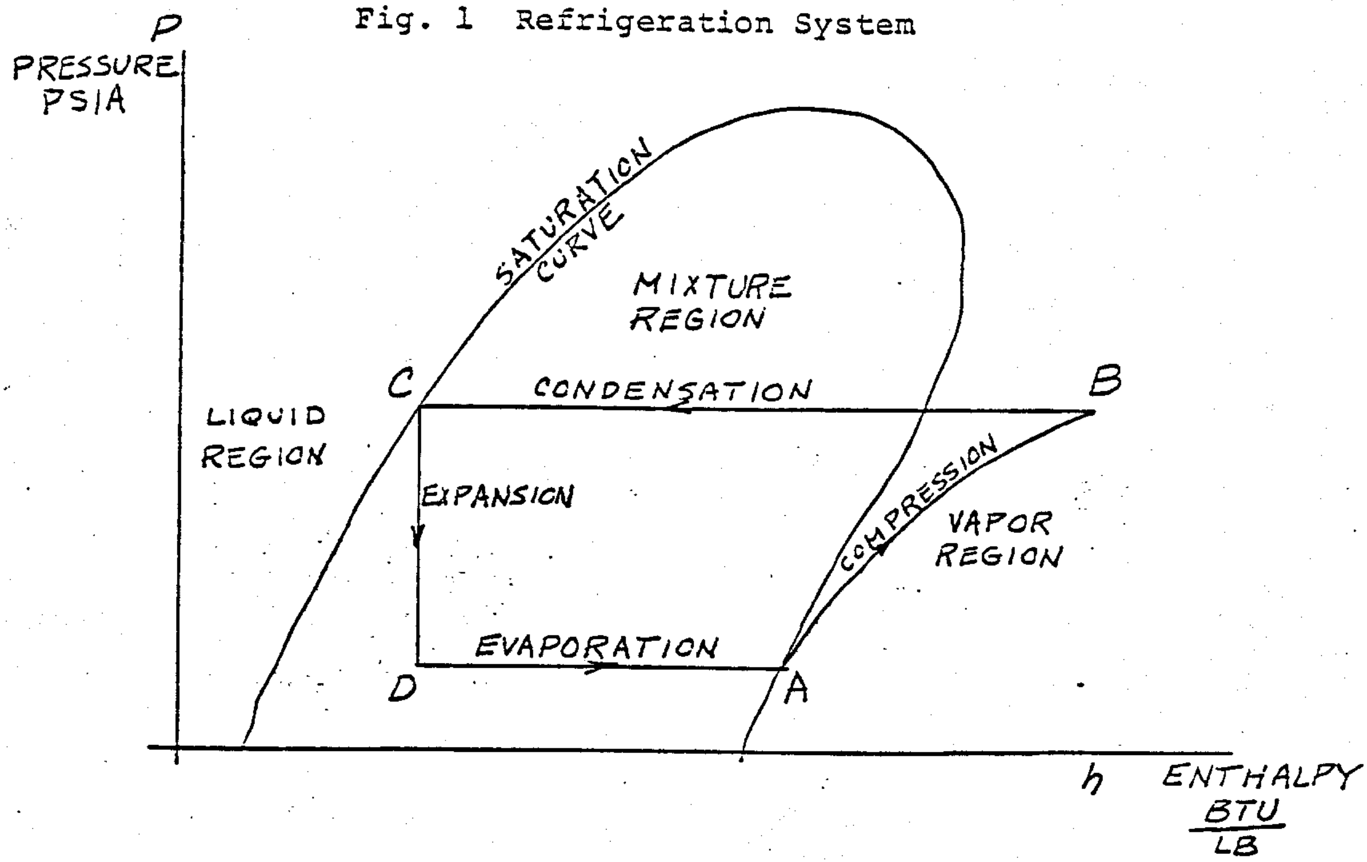


Fig. 2 Pressure - Enthalpy Chart

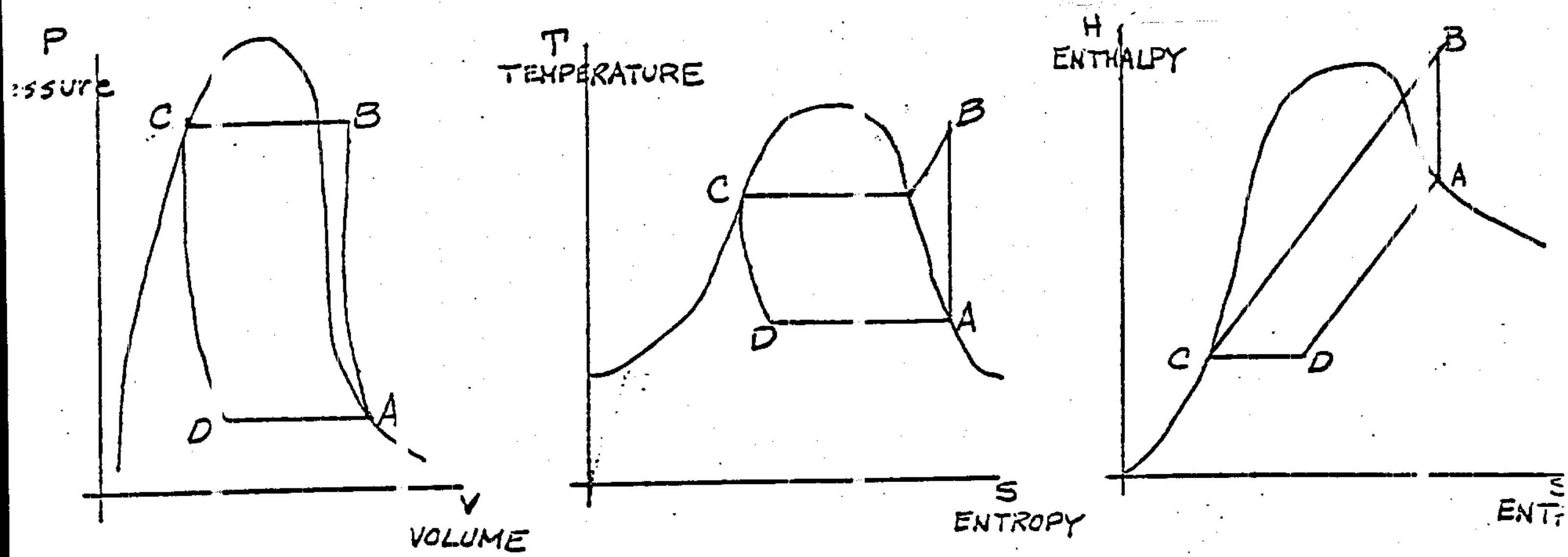


Fig. 3 Refrigeration Cycle on Other Commonly-Used Charts

First Law of Thermodynamics (expressed in Btu/lbm)

$$Q + h + \frac{V_1^2}{2g_oJ} + \frac{Z_1 g}{g_oJ} = W_x + h_2 + \frac{V_2^2}{2g_oJ} + \frac{Z_2 g}{g_oJ}$$

Omitting Kinetic and Potential Energy Terms

For Compressor (neglecting heat loss)

$$h_A = W_x + h_B \quad \text{or} \quad W_x = h_A - h_B$$

$$\text{or} \quad \dot{W}_x \frac{BTU}{hr} = \dot{w} \frac{lb}{hr}\left(h_A - h_B\right)\frac{Btu}{lb}$$

For Condenser

$$Q + h_B = h_c \quad \text{or} \quad Q = h_c - h_B$$

$$\text{or} \quad \dot{Q} \frac{BTU}{hr} = \dot{m} \frac{lb}{hr}\left(h_c - h_B\right)\frac{BTU}{lb}$$

For Evaporator

$$Q + h_D = h_A \quad \text{or} \quad Q = h_A - h_D$$

$$\text{or} \quad \dot{Q} \frac{Btu}{hr} = \dot{m} \frac{lb}{hr}\left(h_A - h_D\right)\frac{Btu}{lb.}$$

If the refrigerating capacity of the system is known and the conditions of the refrigerant entering and leaving the evaporator are known the mass flow rate of the refrigerant is found by

$$\dot{m} = \frac{\dot{Q}_{evap}}{h_A - h_D}$$

This flow rate is the same for all components of the system. The condenser heat rejection can be found by

$$\dot{Q}_{COND} = \dot{m}\,(h_C - h_B)$$

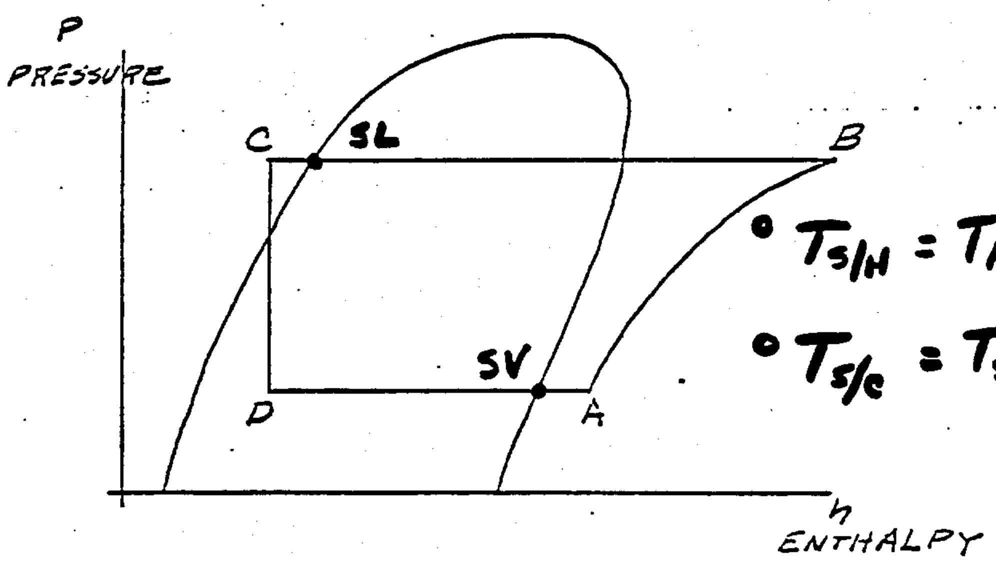
The compressor horse power is found by

$$\dot{W} = \dot{m}\,(h_A - h_B) \text{ times appropriate conversion factor}$$

Definitions:

Degrees of superheat-The number of degrees above saturation temperature.

Degrees of Subcooling-The number of degrees below saturation temperature.



- $T_{S/H} = T_A - T_{SV}$
- $T_{S/C} = T_{SL} - T_C$

Fig. 4 Practical Refrigeration Cycle

Because constant temperature lines are almost vertical in the liquid region the enthalpy of subcooled refrigerant can be found by using saturated liquid enthalpy values. at the refrigerant temperature.

A Freon-12 vapor compression refrigeration system has a compressor of 80% adiabatic efficiency. At inlet to the compressor, the refrigerant is at a pressure of 5.3 psig and has 40 degrees superheat; at discharge from the compressor, the pressure is 100 psia.

Saturated liquid Freon at 100 psia leaves the condenser and passes through an expansion valve into the evaporator coils. The condenser is supplied with cooling water which enters at 60 degrees F and which leaves at 70 degrees F. For a plant capacity of five tons, find:

a. the horsepower required to drive the compressor if it has a mechanical efficiency of 92%.

b. The required capacity in gpm of the cooling water pump.

From Faires: Thermodynamics

Adiabatic Compression Efficiency =

$$\frac{\text{Theoretical Power (isentropic compression)}}{\text{Actual Indicated Power}}$$

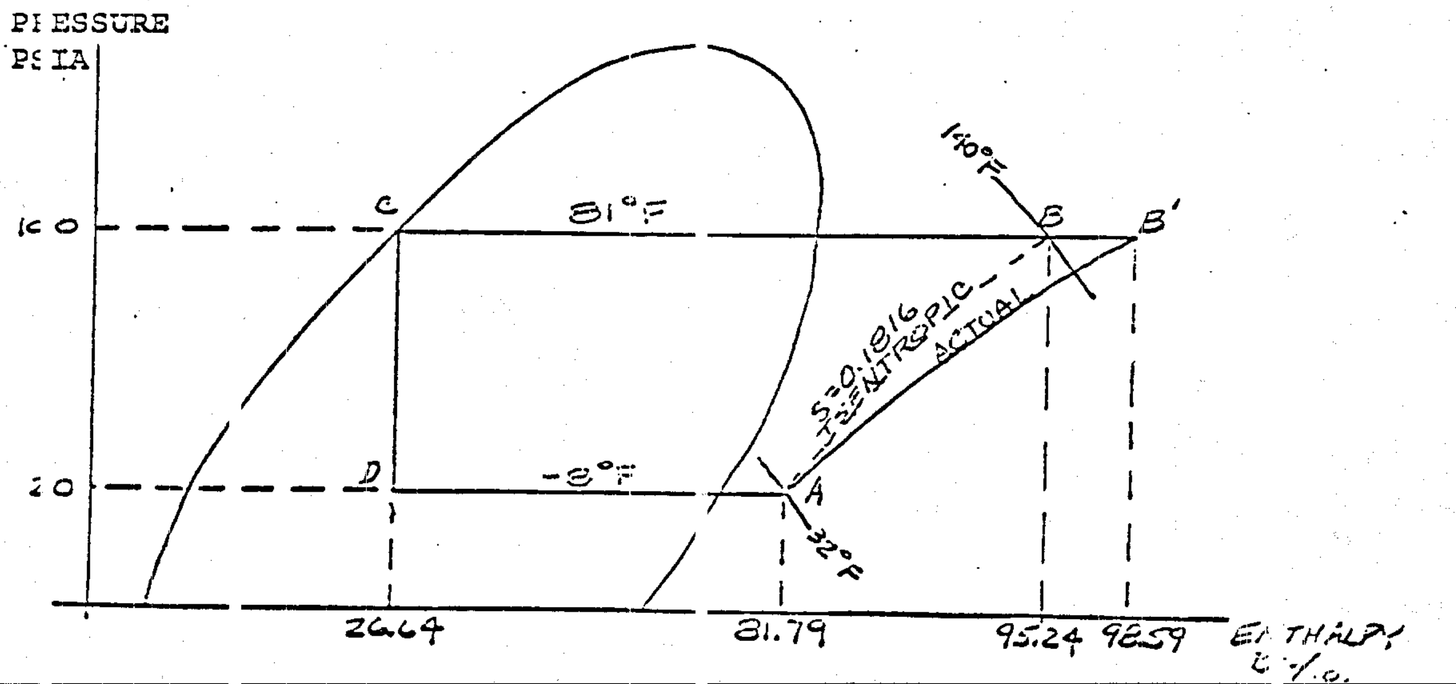
$$0.80 = \frac{h_B - h_A}{h_{B'} - h_A}$$

$$h_{B'} = \frac{h_B - h_A}{0.80} + h_A$$

$$= \frac{95.24 - 81.79}{0.80} + 81.79$$

$$= 98.59 \ \frac{Btu}{lb.}$$

5.3 psig = (14.7 + 5.3) psia = 20 psia



a. Mass flow rate for 5 tons of refrigeration

$$\frac{5 \times 200}{h_A - h_D} = \frac{1000}{81.79 - 26.64} = 18.15 \frac{lb}{min.}$$

$$.02358$$

$$\frac{LB}{MIN} \times \frac{BTU}{LB} \times \left(\frac{778 \times}{60 \times 550} \right)$$

Theoretical Compressor Horsepower

$$\text{Theor. H.P.} = \dot{m}(h_{B'} - h_A) \times 0.02356$$
$$= 18.15(98.59 - 81.79) \times 0.02356$$
$$= 7.19 \text{ H.P}$$

Actual Horsepower = $\frac{\text{THEORETICAL H.P.}}{\text{EFFICIENCY}} = \frac{7.19}{0.92} = 7.80 \text{ H.P.}$

H Heat Rejection of Condenser

$$\dot{Q}_{COND} = \dot{m}(h_{B'} - h_c)$$
$$= 18.15(98.59 - 26.64)$$
$$= 18.15 \times 71.95$$
$$= 1,308 \frac{Btu}{min}$$

By the sensible heat relationship

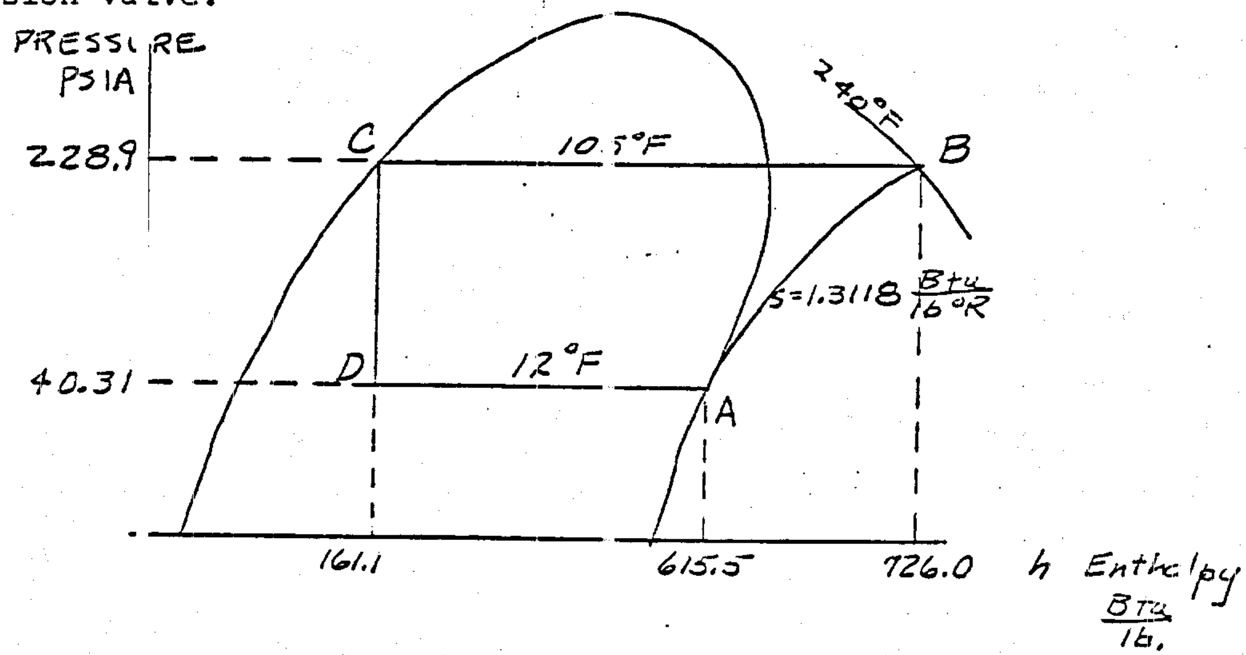
$$\dot{Q} = \dot{m}\, c\, \Delta t$$

$$\dot{m} = \frac{\dot{Q}}{c \Delta t} = \frac{1,308}{1.0 \times 10} = 130.8 \frac{lb.}{min.}$$

$$= \frac{130.8}{8.33} = 15.7 \frac{gal.}{min.}$$

$$.02356 = \frac{778}{60 \times 550}$$

An ammonia refrigeration unit has an 180-rpm, double-acting compressor whose cylinder has a diameter of 8 in. with a 10-inch stroke and whose clearance is 5%. It is desired to calculate the capacity of the unit in tons of refrigeration at an evaporation temperature of 12°F and a condensing temperature of 105°F. Assume that the 12°F vapor is saturated as it enters the compressor. Also assume that the liquid ammonia is not subcooled at the entrance to the expansion valve.



Theoretical Displacement = $2 \times \frac{\pi\left(\frac{8}{12}\right)^2}{4} \times \frac{10}{12} \times 180 = 104.9\ \frac{ft.^3}{min}$

From: APPLICATION OF THERMODYNAMICS by BERNARD D. WOOD

Volumetric Efficiency

$$\eta_v = 1 + CF - CF\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$ THEORETICAL VOLUMETRIC EFF.

where CF = $\frac{\text{CLEARANCE VOLUME}}{\text{PISTON DISPLACEMENT}}$

From Mark's Handbook K for ammonia = 1.29 K $\cong$ n for this process

Clearance = CF = 0.05

$$\eta_v = 1 + 0.05 - 0.05\left(\frac{228.9}{40.31}\right)^{\frac{1}{1.29}}$$

$$= 1.05 - 0.05 \times 5.68^{0.775}$$

$$= 1.05 - 0.05 \times 3.84$$

$$= 1.05 - 0.192$$

$$= 0.858$$

The actual volumetric flow rate is (handwritten insertion: THEORETICAL)

$$104.9 \times 0.858 = 89.9 \frac{ft^3}{min}$$

The mass flow rate is

$$\dot{m} = \frac{\dot{V}}{v} = \frac{89.9}{6.996} = 12.82 \frac{lb.}{min.}$$

The evaporator capacity is

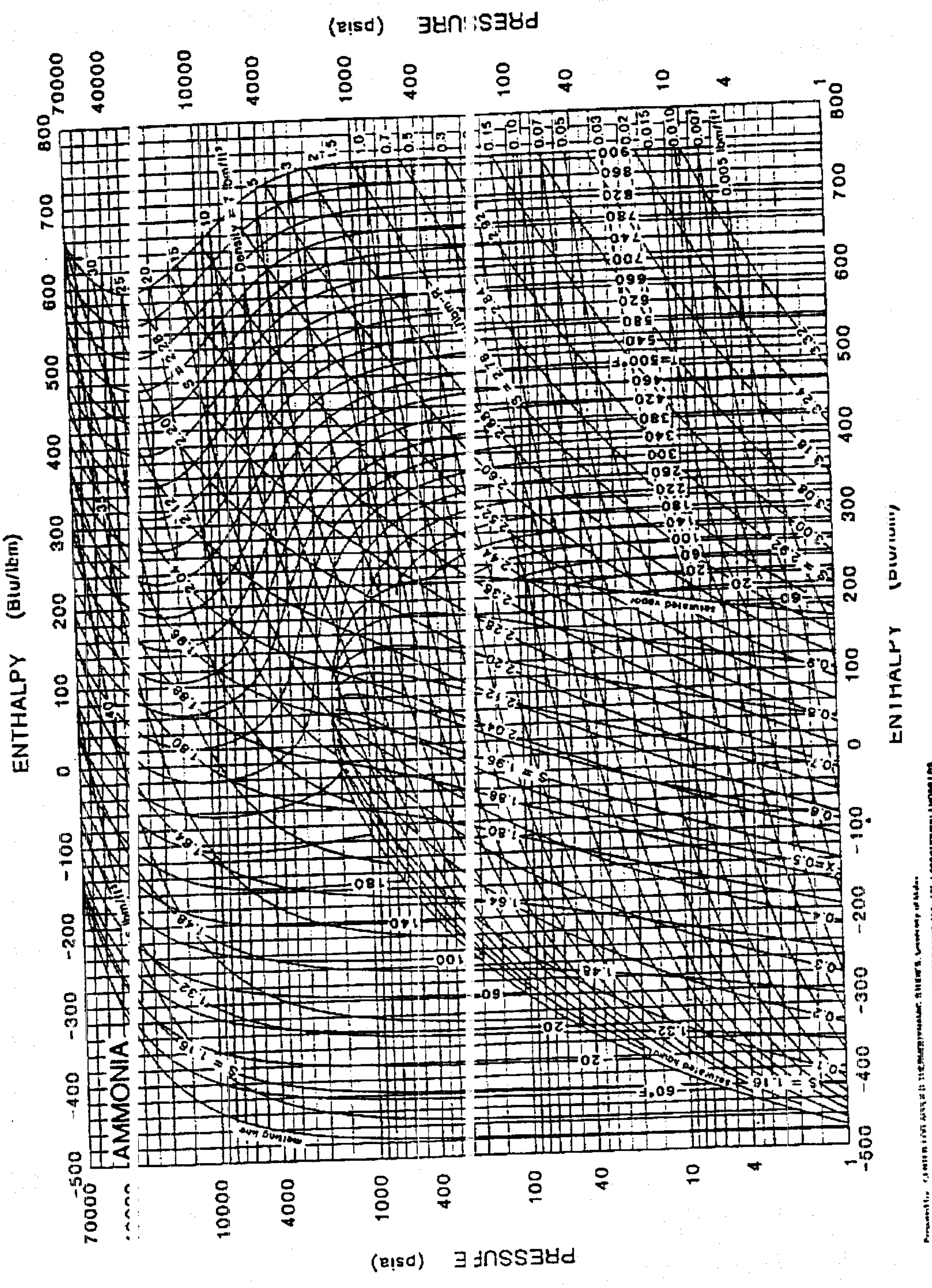
$$\dot{Q} = \dot{m}(h_A - h_D)$$

$$= 12.82(615.5 - 161.1)$$

$$= 12.82 \times 454.4$$

$$= 5,830 \frac{BTU}{min}$$

$$\dot{Q} = \frac{5,830}{200} = 29.1 \text{ Tons}$$


AMMONIA
ENTHALPY (Btu/lbm)
PRESSURE (psia)
saturated liquid
saturated vapor
melting line
Density = 7 lbm/ft³
T=500°F

(NCEE April 1971)

A given refrigeration system uses Freon-12 for the working medium and produces 50,000 Btu/hr cooling with the following operating conditions:

The compressor discharge pressure and temperature are 125 psia and $140^{\circ}F$, respectively.

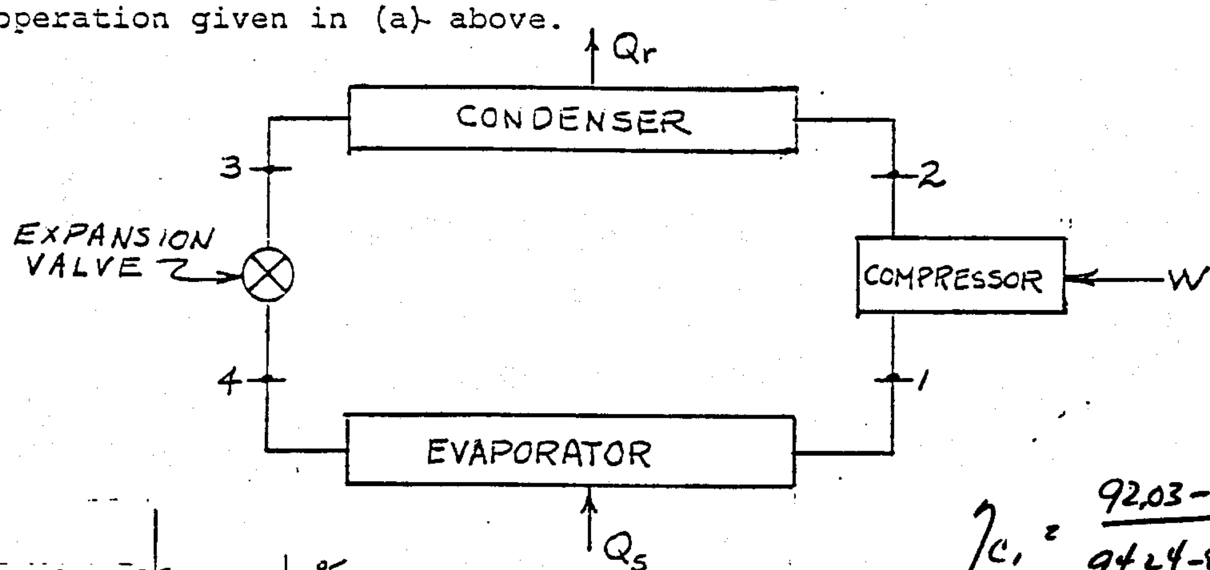
The temperature of the Freon leaving the condenser is $90^{\circ}F$.

The pressure and temperature leaving the evaporator are 50 psia and $60^{\circ}F$, respectively.

(a) If the system has a Positive Displacement constant speed compressor, and the cooling area conditions are changed to necessitate operation at a pressure of 35 psia and a temperature of $40^{\circ}F$ in the evaporator, determine (by use of either chart or table values) the expected cooling capacity at the new condition of operation. Assume that the temperature of the Freon leaving the condenser and the discharge pressure leaving the compressor remain unchanged. (Please use station numbers as shown on sketch).

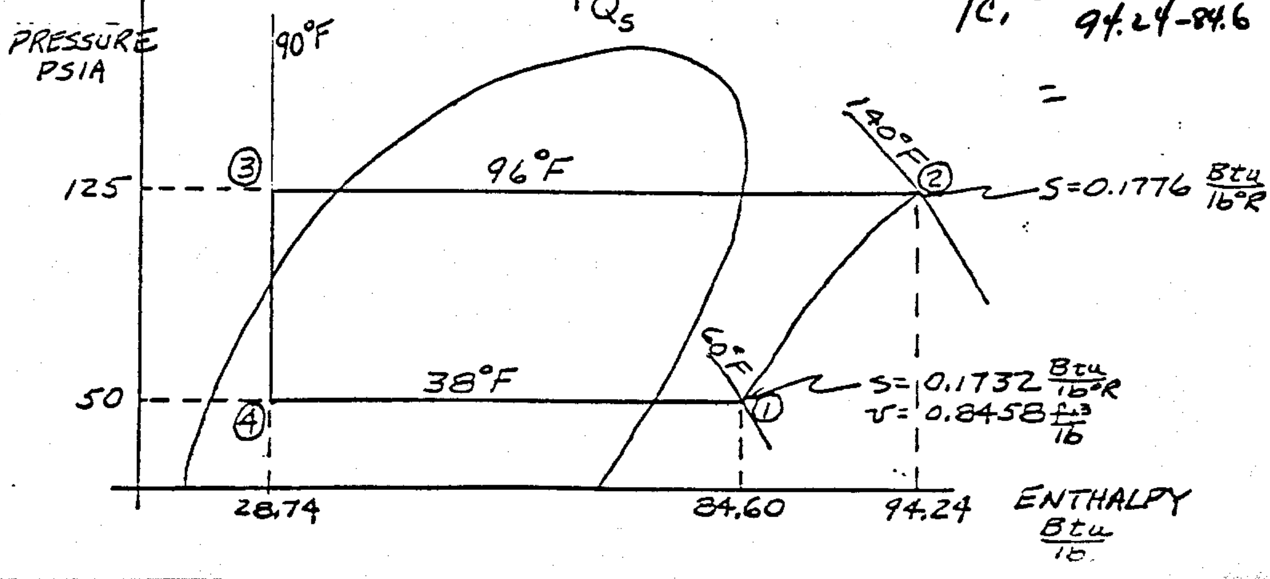
~ SAME AS η_c = CONST

(b) If the compression process is adiabatic and the rate of entropy increase remains essentially the same during compression, determine by calculations the C.O.P. for the original condition of operation and for the condition of operation given in (a) above.



$$\eta_{c,} = \frac{92.03-84.6}{94.24-84.6} =$$

$=$



ASS Flow Rate:

$$\dot m = \frac{\dot Q}{h_1 - h_4} = \frac{50,000}{84.60-28.74}$$

$$= \frac{50,000}{55.86} = 894 \frac{lb.}{hr.}$$

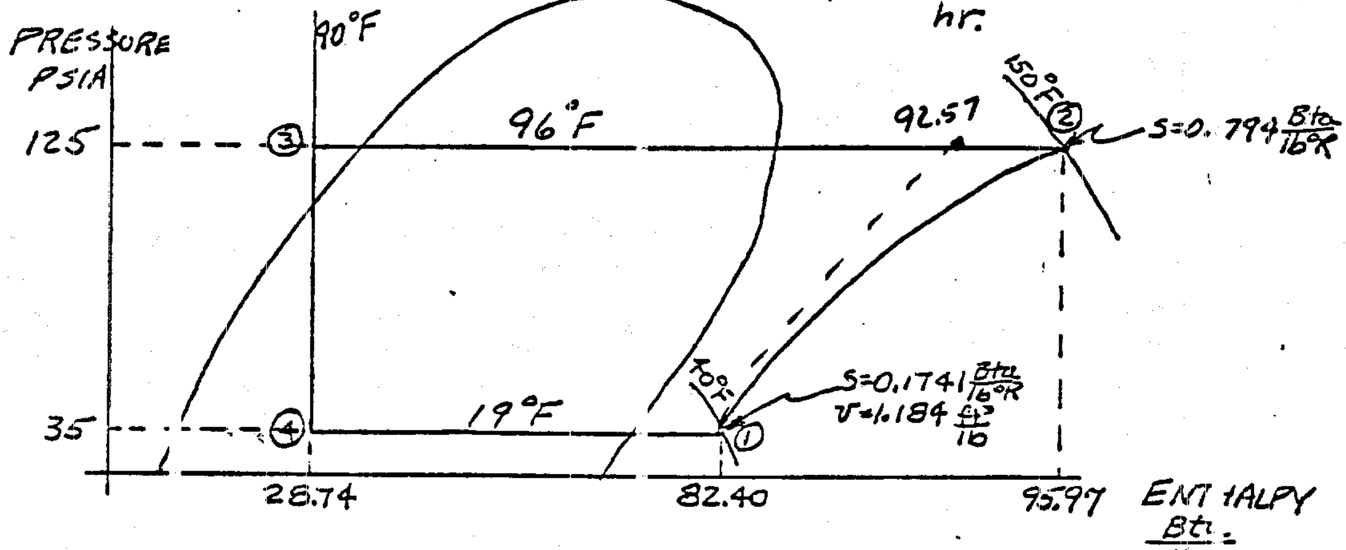
Volume Flow Rate:

$$\dot V \frac{ft^3}{hr} = \dot m \frac{lb.}{hr.} \cdot v_1 \frac{ft^3}{lb}$$

$$= 894 \cdot 0.8458$$

$$= 756 \frac{ft^3}{hr.}$$

art (A)



e volume flow rate s the same as the initial condition 756 ft^3/hr.

ss Flow Ra:e:

$$\dot m \frac{lb}{hr} = \frac{\dot V \frac{ft^3}{hr}}{v_1 \frac{ft^3}{lb}}$$

(ACTUALLY LESS DUE TO HIGHER PRESSURE RATIO AND LOWER η_{vol})

BUT C_L IS UNKNOWN

COULD ESTIMATE USING ASSUM C_L

$$\dot m = \frac{756}{1.184} = 639 \frac{lb.}{hr.}$$

oling Capacity:

$$\dot Q \frac{Btu}{hr} = \dot m \frac{lb}{hr} (h_1 - h_4) \frac{Btu}{lb}$$

$$= 639 (82.40 - 28.74)$$

$$= 639 \cdot 53.66$$

$$= 34,300 \frac{Btu}{hr.}$$

t (B)

ding rate of entropy increase at initial conditions:

$$S_1 = 0.1732 \frac{Btu}{lb.°R} \qquad S_2 = 0.1776 \frac{Btu}{lb\,°R}$$

$$\Delta S = 0.0044 \frac{Btu}{lb\,°R}$$

e of entropy increase $= \frac{\Delta S}{\Delta P} = \frac{0.0044}{75} = 5.88 \times 10^{-5} \frac{Btu}{lb\,°R\,psi}$

Final entropy for part (a)

$$\Delta S = Rate \times \Delta P$$
$$= 5.88 \times 10^{-5} \times 90$$
$$= 5.3 \times 10^{-3}$$
$$= 0.0053 \ \frac{Btu}{lb\,°R}$$

$$S_2 = S_1 + \Delta S$$
$$= 0.1741 + 0.0053$$
$$= 0.1794 \ \frac{Btu}{lb\,°R}$$

C.O.P. for initial conditions:

$$C.O.P. = \frac{84.60 - 28.74}{94.24 - 84.60} = \frac{55.86}{9.64} = 5.78$$

C.O.P. for part (A):

$$C.O.P. = \frac{82.40 - 28.74}{95.97 - 82.40} = \frac{53.66}{13.57} = 3.96$$

OR
BASED ON COMPR. EFF ~~[illegible]~~

$$\eta_{T_1} = \eta_{T_2}$$

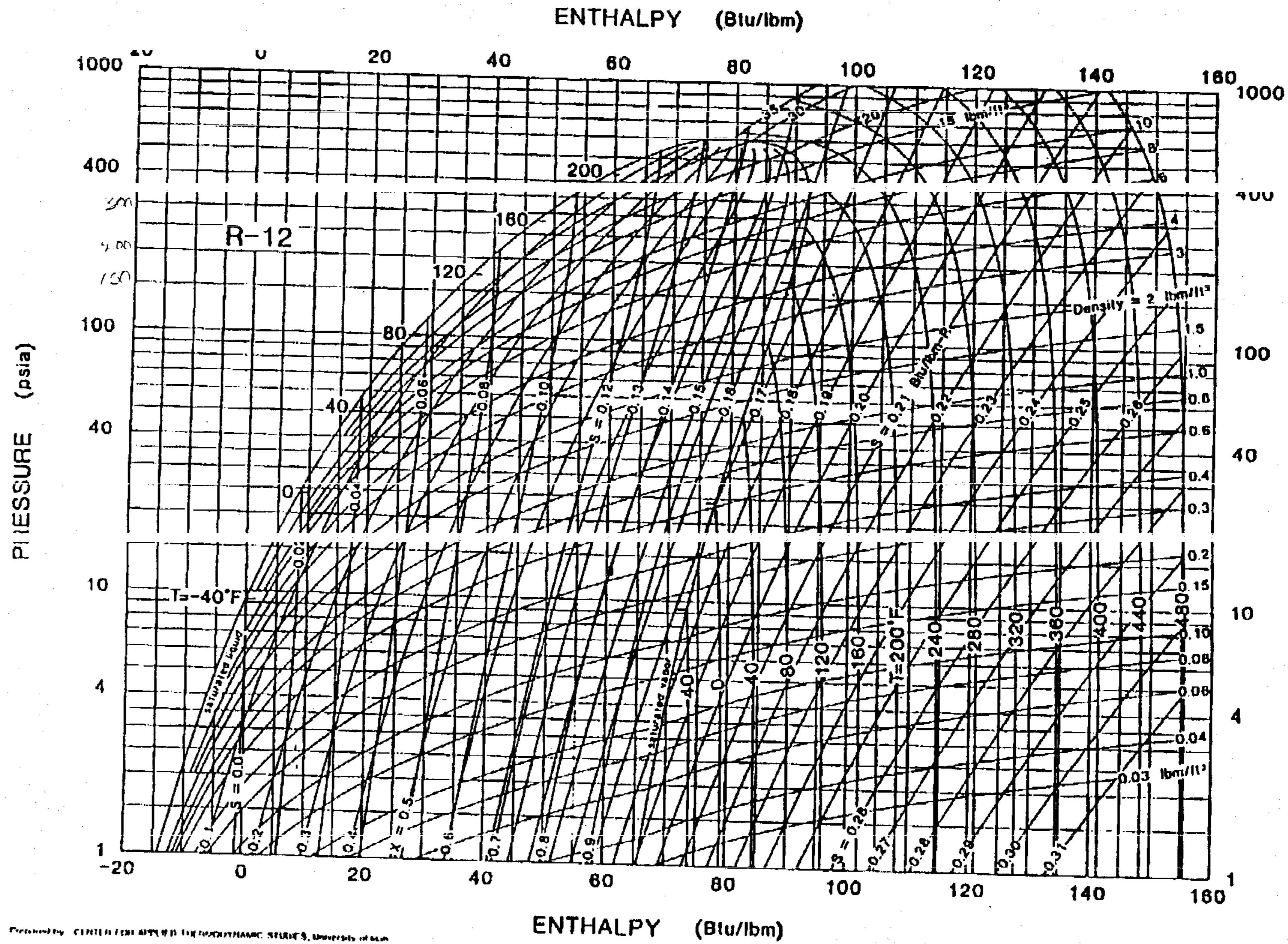


Fig. 2 Pressure-Enthalpy Diagram for Refrigerant 12

ADDENDUM TO PROBLEM NCEE APRIL 1971

$$K = \frac{C_p}{C_v} = \frac{.1516}{.1305} = 1.16$$

ASSUMING $C_L = .05$

$$\eta_{V_1} = 1 + C_L - C_L \left(\frac{P_2}{P_1}\right)^{1/k} = 1.05 - .05\left(\frac{125}{50}\right)^{1/1.16} = .94$$

$$\eta_{V_2} = 1.05 - .05\left(\frac{125}{35}\right)^{1/1.16} = .90$$

ADJUSTING FOR CHANGE IN η_V:

$\therefore$ NEW COOLING CAPITY $= 34300 \times \frac{.90}{.94} = 32400$ BTU/HR

USING ADIABATIC COMPRESSOR EFFICIENCY INSTEAD OF ENTROPY CHANGE RATE:

$$\eta_{C_1} = \frac{92.03 - 84.6}{94.24 - 84.6} = .771$$

ASSUME η_C = CONST i.e. $\eta_{C_1} = \eta_{C_2}$

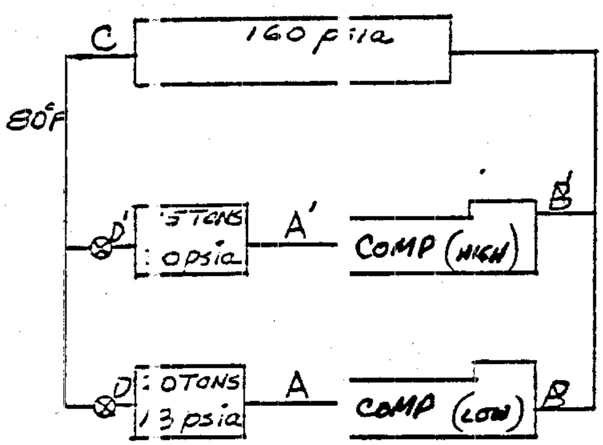
$$\therefore h_2 = 82.4 + \frac{92.57 - 82.4}{.771} = 95.6 \; (96)$$

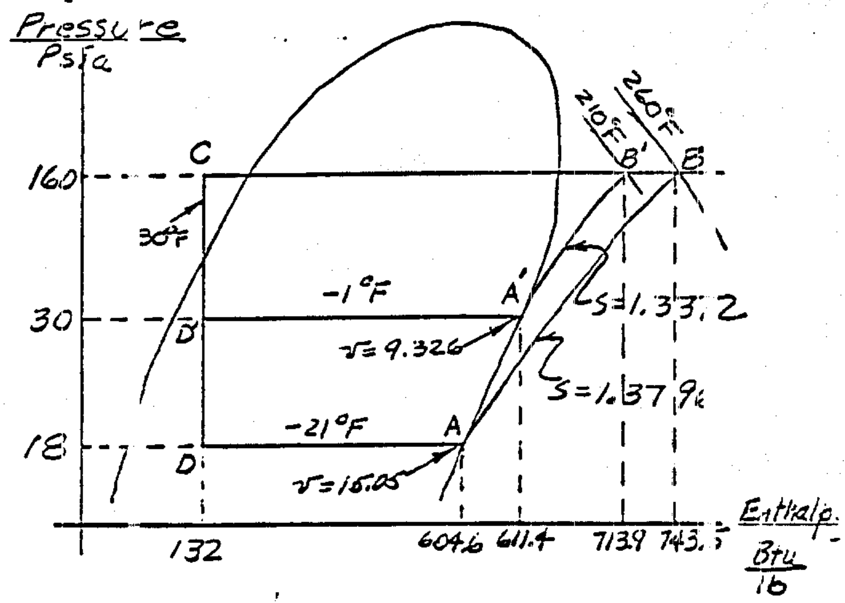
REFRIGERATION PROBLEM FROM APRIL, 1969 P. E. EXAM

A packing plant using ammonia as the refrigerant requires 15 tons of refrigeration at an evaporator pressure of 30 psia, and 20 tons at 18 psia. Each evaporator, by means of its own compressor, discharges to a common condenser operating at a pressure of 160 psia. The ammonia leaving the condenser is subcooled to 80 F, after which it passes through separate expansion valves to the evaporators.

Using an ideal vapor-compression cycle, and assuming saturated vapor at the inlet of each compressor, find:

(a) The overall horsepower required per ton of refrigeration
(b) The displacement of each compressor.





Low Pressure System

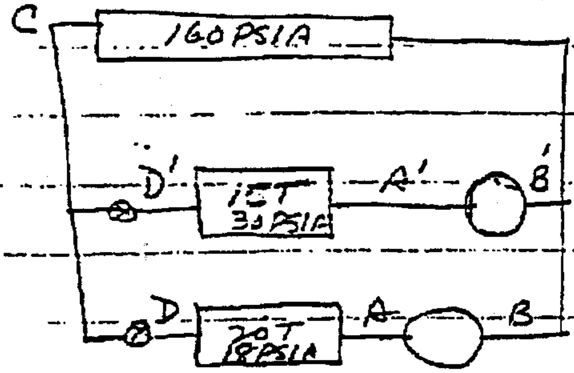
Evaporator $\dot{Q}_{\frac{Btu}{min}} = \dot{m}_{\frac{lb}{min}} (h_A - h_D)_{\frac{Btu}{lb}}$

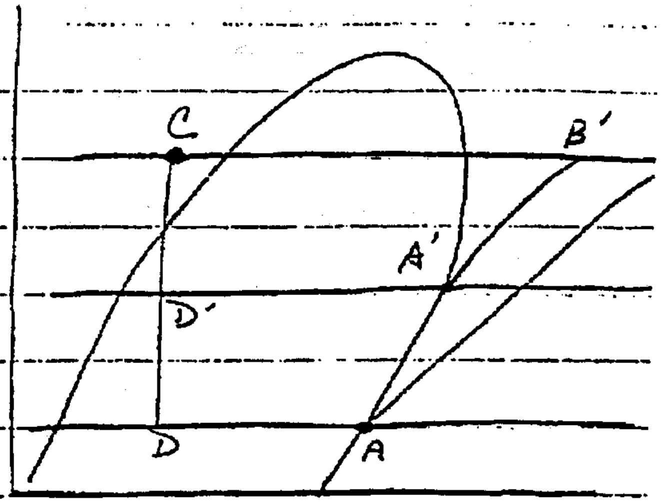
$$\dot{m} = \frac{\dot{Q}}{h_A - h_D} = \frac{20 \times 200}{604.6 - 132} = \frac{4000}{472.6} = 8.45 \frac{lb}{min}$$

Compressor $\dot{W}_{x\,\frac{Btu}{min}} = \dot{m}_{\frac{lb}{min}} (h_A - h_B)_{\frac{Btu}{lb}}$

$$\dot{W}_{x\,\frac{Btu}{min}} = 8.45(604.6 - 743.5) = 8.45 \times (-138.9) = -1171 \frac{Btu}{min}$$

$$H.P. = 1171 \frac{Btu}{min} \times 2.356 \times 10^{-2} \frac{HP}{Btu/min} = 27.6\ HP$$





POINT	P	T	S	h	v
A	18	−21	2.56	200	14.9
A'	30	−1	2.52	205	9.23
B	160	260	2.56	343	
B'	160	213	2.52	305	
C	160	80		−275	
D	18	−21		−275	
D'	30	−1		−275	

LOW PRESS. SYSTEM

$Q_{EVAP} = \dot{m}(h_A - h_D)$

$\dot{m} = \frac{20 \times 200}{200 - (-275)} = 8.4 \ LB/MIN$

COMPRESSOR $\dot{W} = \dot{m}(h_A - h_B)$

$= 8.4\,(343 - 200) = 1204 \ BTU/MIN$

$= 1204 \times .02356 = 28.4 \ HP$

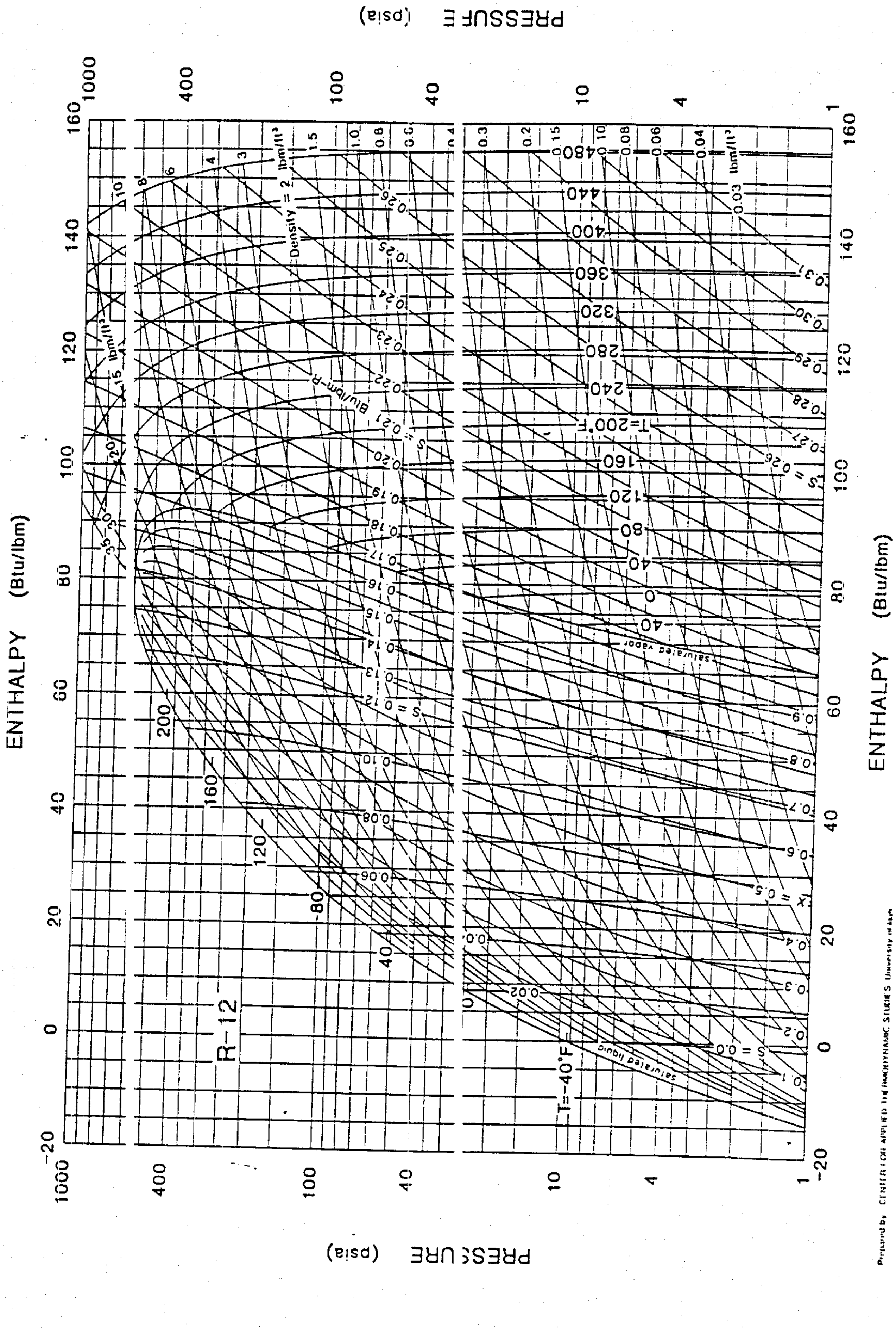
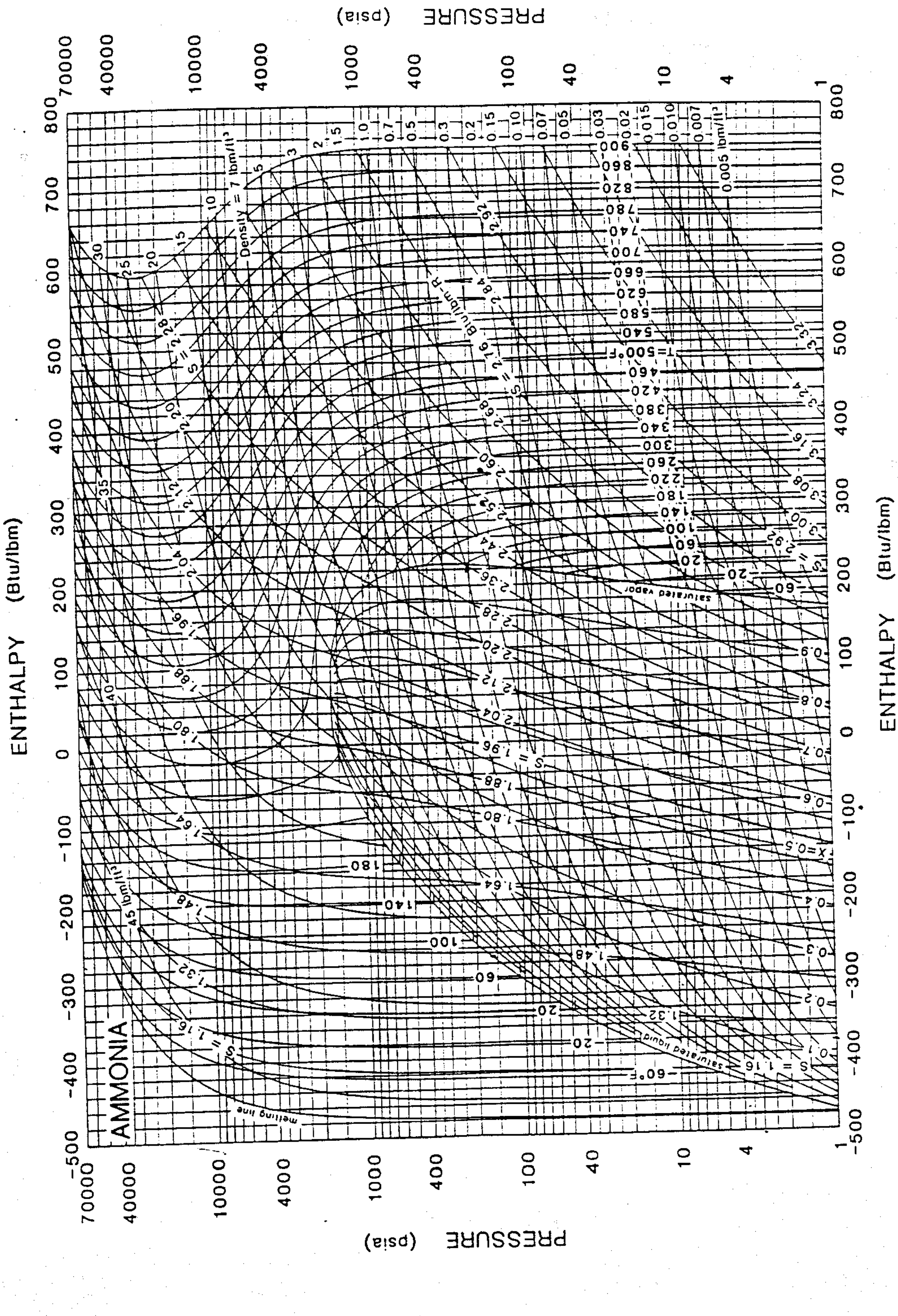


Fig. 2 Pressure-Enthalpy Diagram for Refrigerant 12


AMMONIA
ENTHALPY (Btu/lbm)
PRESSURE (psia)
saturated liquid
saturated vapor
melting line
Density = 7 lbm/ft³
45 lbm/ft³
0.005 lbm/ft³
T = 500°F
S = 2.16 Btu/lbm-R

HIGH PRESS. SYSTEM

EVAP $\dot{m} = \frac{15 \times 200}{205 - (-275)} = 6.3$ LB/MIN

COMPR. $\dot{W} = 6.3 \times (205 - 305) = 630$ BTU/MIN

$= 14.7$ HP

OVERALL HP/TON $= \frac{28.4 + 14.7}{20 + 15} = 1.23$

COMPRESS. DISPL.

LOW $\dot{V} = 8.4 \times 14.9 = 125.2$ CFM

HIGH $\dot{V} = 6.3 \times 9.23 = 58.1$ CFM

A refrigeration system is to be designed to produce a cooling load of 20 tons of refrigeration. The working medium is to be Freon F-12. The pressure in the evaporator is 15 psia and the temperature leaving the evaporator is 20°F. The temperature leaving the compressor is 180°F and the pressure is 150 psia. For the simple cycle, the conditions entering the compressor are the same as those leaving the evaporator. Neglect losses for both heat transfer and pressure drop in the lines and for pressure drop in the condenser and evaporator. The temperature leaving the condenser is 100°F. Calculate the following for the simple cycle:

(a) The required mass rate of Freon flow in lbm/min.

(b) The required compressor displacement for a volumetric efficiency of 80%. Give answer in cfm.

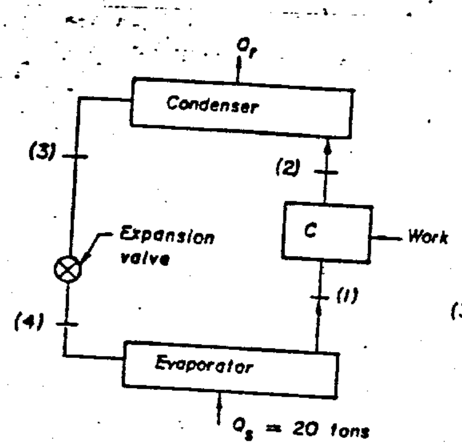
If a regenerative heat exchanger can be used, as illustrated in the sketch, to increase the temperature entering the compressor to 40°F with no change in conditions leaving the evaporator or condenser, neglect losses and pressure drop and find:

(c) The new mass rate of Freon flow required in lbm/min.

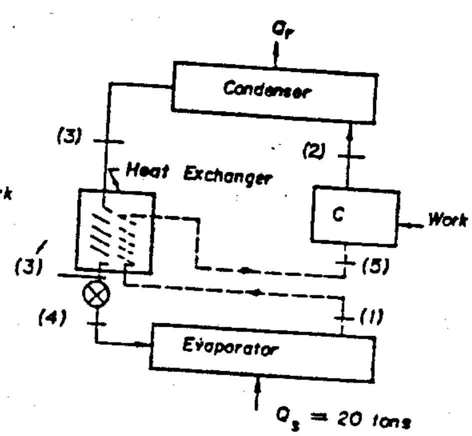
(d) The new required compressor displacement at the same 80% volumetric efficiency. Give answer in cfm.

Please use same station numbers as indicated on the sketch.

State reference used for table or chart values.

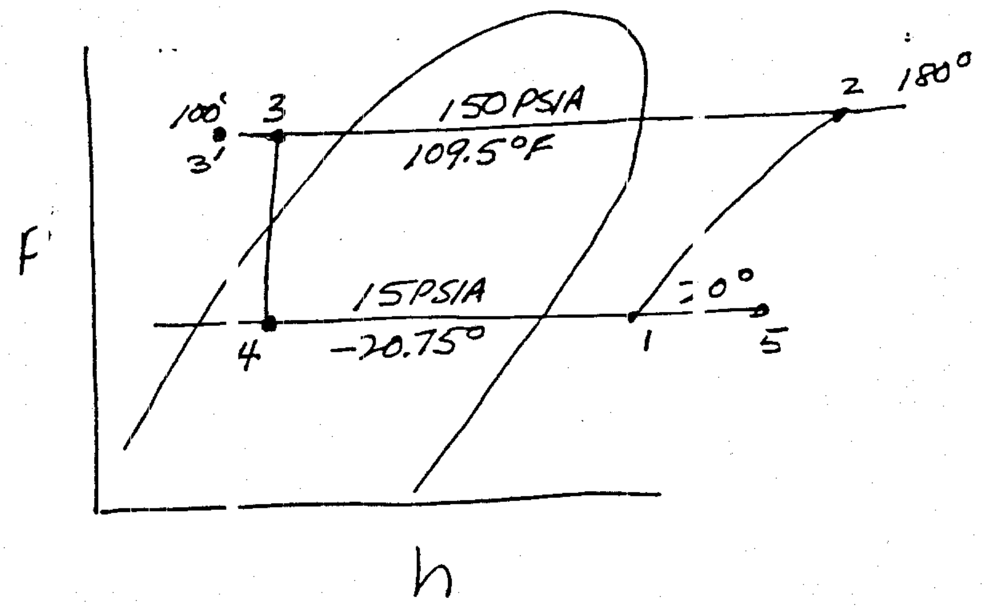


Simple Cycle



Regeneration Cycle

PRoB # 146 NCEE 72-76



$h_1 = 80.71$ $S_1 = .1835$ $v_1 = 2.7494$ FT³/LB

$h_2 = 100.73$ $S_2 = .1845$ $v_2 = .332$ F³/LB

$h_3 = h_4 = 31.22$ (EVALUATED @ 100° SAT.)

a. $Q_{REFRIG} = \dot{m}(h_1 - h_3)$

$$\dot{m} = \frac{20 \times 200}{80.71 - 31.22} = \frac{4000}{49.49} = 80.8 \text{ LB/MIN}$$

b $\dot{V}_{REQ} = \dot{m} v = 2.7494 \times 80.8 = 222 \ FT^3/MIN$

$\dot{V}_{ACT} = \frac{222}{\eta_V} = \frac{222}{.8} = 277.8$ CFM

$\eta_V = \frac{\dot{m}_{ACT}}{\dot{m}_{THEOR}} = \frac{\dot{V}_{ACT}}{\dot{V}_{THEOR}}$

c AT 40°F & 15 PSIA

$h_5 = 83.56$ BTU/LB $\quad v_5 = 2.877 \ FT^3/LB$

$\therefore h_{2'} = h_3 - (h_5 - h_1)$ ENERGY BAL. ON HEX

$= 31.22 - (83.56 - 80.71) = 28.37$

$\dot{m} = \frac{20 \times 200}{80.71 - 28.37} = 76.4$ LB/MIN

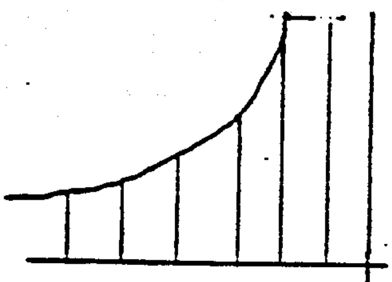
NOTE: h_1 NOT h_5

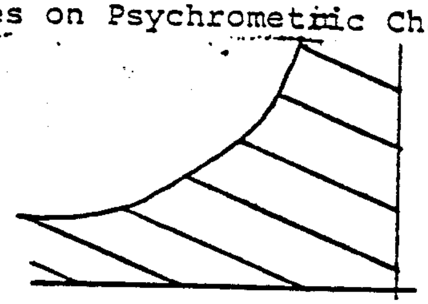
d $\dot{V}_{ACT} = \frac{76.4 \times 2.877}{.8} = 274.8$ CFM

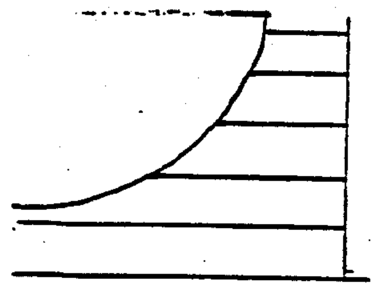
Definitions:

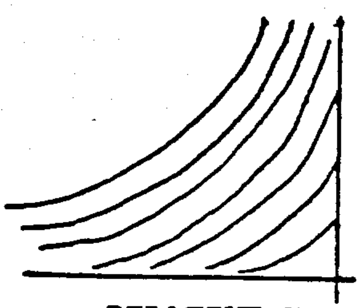
Dry Bulb Temperature - The temperature of a mixture of gasses indicated by an accurate thermometer.
Wet Bulb Temperature - Temperature at which liquid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature
Adiabatic Process - No net heat is added to or taken from a substance or system undergoing the process.
Dew Point Temperature - Temperature at which condensation will just begin when the moist air mixture under consideration is cooled at constant pressure
Humidity Ratio or Specific Humidity - The mass of water vapor per unit mass of dry air in a vapor-air mixture GRAINS H_2O/LB D.A. 7000 grains = 1 pound
Relative Humidity - The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.
Sensible Heat Load - Heat added or removed that changes only the temperature of the air vapor mixture.
Latent Heat Load - Heat required to evaporate water vapor into or remove water vapor from the air vapor mixture.
Sensible Heat Ratio - Ratio of the sensible heat load to the total heat load of the space considered.
Standard Air - Air with a density of 0.075 lb. per cubic foot. This is substantially equivalent to dry air at 70°F and 29.92 in.Hg. barometer.

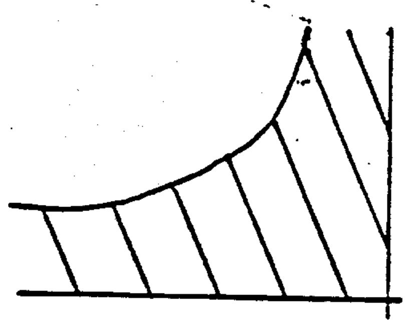
Lines on Psychrometric Chart

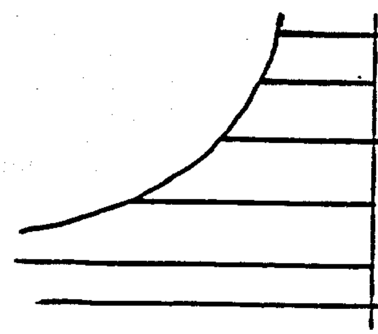

DRY BULB

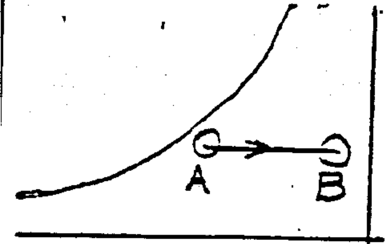

WET BULB


DEW POINT

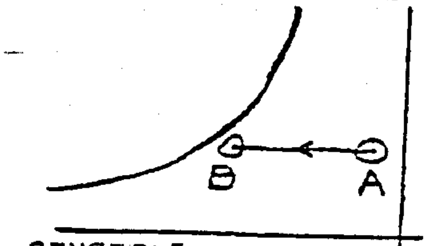

RELATIVE HUMIDITY


SPECIFIC VOLUME

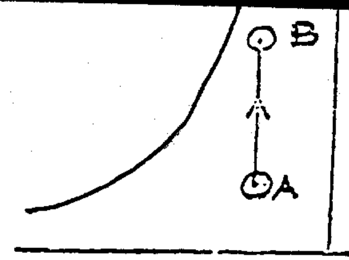

HUMIDITY RATIO



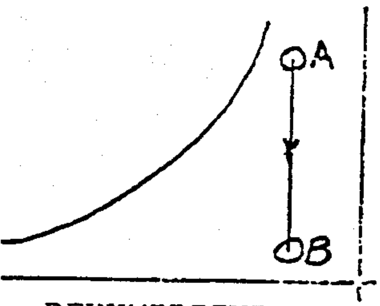
SENSIBLE HEATING



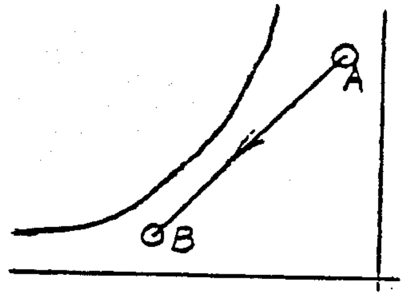
SENSIBLE COOLING



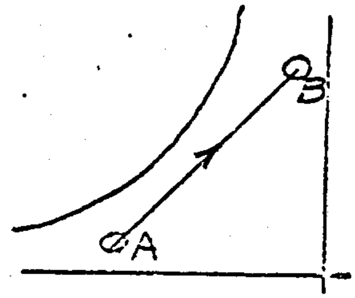
HUMIDIFYING



DEHUMIDIFYING



COOLING AND DEHUMIDIFYING



HEATING & HUMIDIFYING

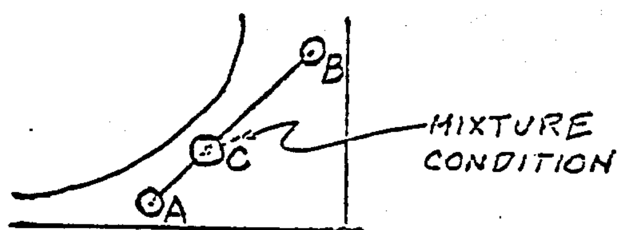
AIR MIXTURES

The resulting condition of a mixture of two air supplies at different conditions can be found on the psychrometric chart. First plot the two conditions on the chart and draw a line between them. The resulting condition will be on this line at the dry bulb calculated below.

$\dfrac{\text{CFM "A" air}}{\text{CFM Total}}$ x D.B. "A" air =

$\dfrac{\text{CFM "B" air}}{\text{CFM Total}}$ x D.B. "B" air =

Sum of numbers = D.B. Mixed Air =



If the three conditions and the total CFM are given the CFM of each component can be found by a ratio of the dry bulb temperatures

$$\text{CFM "A"} = \text{Total CFM} \times \frac{t_B - t_C}{t_B - t_A}$$

$$\text{CFM "B"} = \text{Total CFM} \times \frac{t_C - t_A}{t_B - t_A}$$

U.S. (I.P.) UNITS
AIR CONDITIONING
"MAGIC" NUMBERS

$Q_{SENS} = 1.08 \times CFM \times \Delta T_{SENS}\ (°F)$ BTU/HR

$Q_{LAT.} = .68 \times CFM \times \Delta W \left(\frac{GR}{LB_{DA}}\right)$ BTU/HR

$Q_{TOTAL} = 4.5 \times CFM \times \Delta h$ BTU/HR

$1.08 = 4.5 \times .24$

$\frac{FT^3}{MIN} \times 60\ \frac{MIN}{HR} \times .075\ \frac{LB}{FT^3} = 4.5 \times CFM\ \frac{LB}{HR}$

$.68 = 1060\ \frac{BTU}{LB_{H_2O}} \times \frac{FT^3}{MIN} \times \frac{GR_{H_2O}}{LB_{DA}} \times 60\ \frac{MIN}{HR} \times .075\ \frac{LB}{FT^3} \times \frac{1}{7000}\ \frac{LB}{GR}$

When conditioned air enters a room it mixes with the room air and comes into equillibrium with the heat load of the room. The cool supply air is warmed by the various sources of heat in the room. The resulting equillibrium temperature is the design condition of the room.

The sensible heat ratio $\left(\frac{\text{sensible heat gain}}{\text{total heat gain}}\right)$ is an index of the type of load. If the ratio is high it is a dry type of load (telephone equipment room). If it is low, it is a wet type of load (laundry). To maintain conditions in a room the air supply must be on the sensible heat ratio line.

To draw the sensible heat ratio line thru any point on the chart, line up the reference point with the calculated sensible heat ratio on the scale to the right of the chart. This line has the correct slope. Now, a line parallel to this one and thru the desired condition will give the true sensible heat ratio line. To maintain the desired conditions the supply air must be on this line.

If the dry bulb of the supply conditions is known the CFM required can be found from the sensible heat equation

$$Q_{SENSIBLE} = 1.08\ CFM\,(t_2 - t_1) \quad BTU/HR$$

$$\text{or} \quad CFM = \frac{Q_{SENSIBLE}}{1.08\,(t_2 - t_1)} \qquad\qquad 1.08 = 4.5 \times .24$$

The total load equation uses the change in enthalpy between two conditions.

$$Q_{TOTAL} = 4.5\ CFM\,(h_2 - h_1)$$

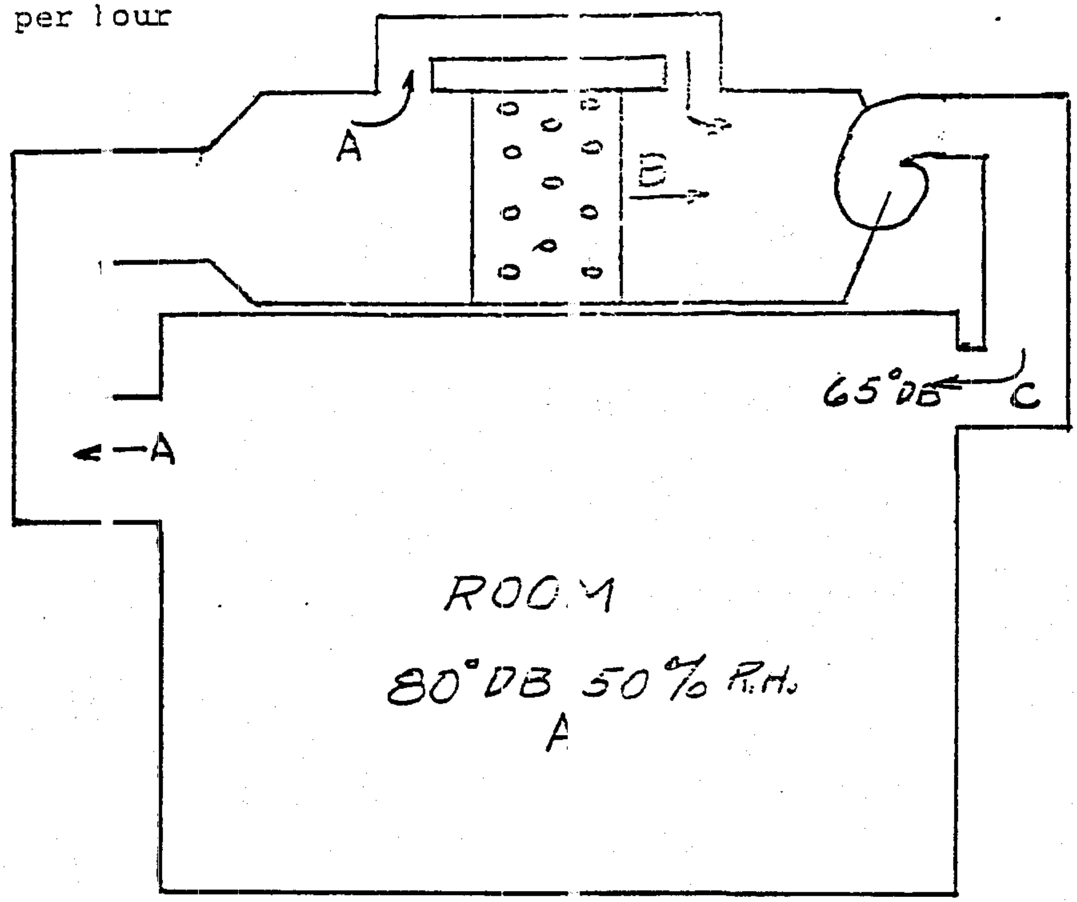
$$\text{or} \quad CFM = \frac{Q_{TOTAL}}{4.5\,(h_2 - h_1)}$$

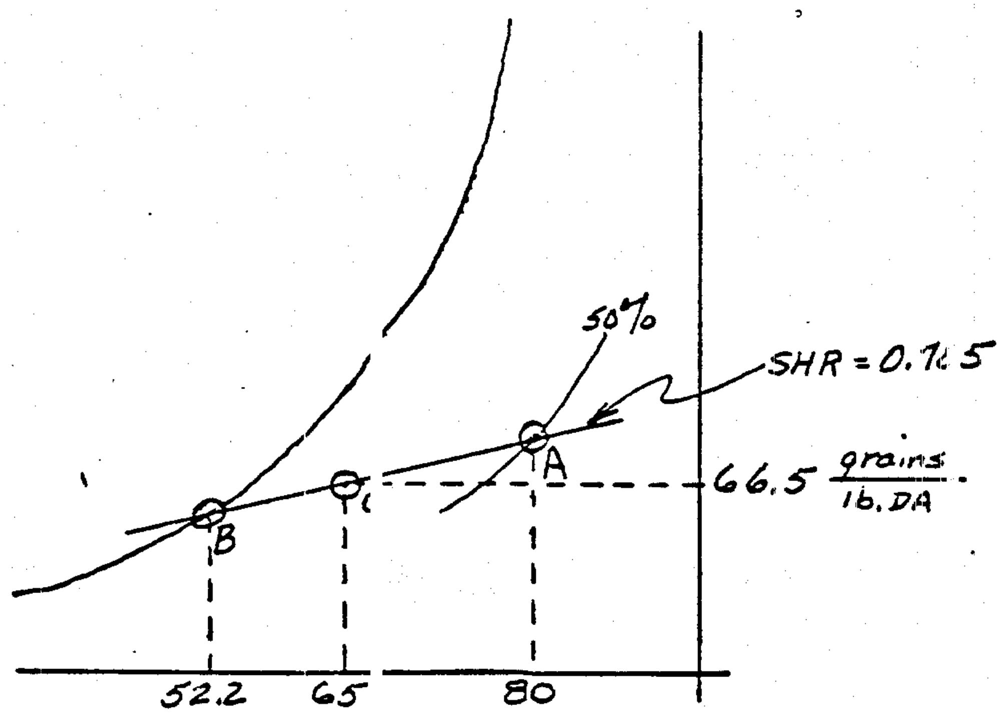
$$\frac{FT^3}{MIN} \times 60\ \frac{MIN}{HR} \times .075\ \frac{LB}{FT^3} = 4.5 \times CFM$$

PROBLEM:

A certain space is to be air-conditioned with supply air entering at 65 dbt. Sensible and latent heat loads are 600,000 and 250,000 Btu/hr, respectively. Air leaving the conditioned space is to be at 80 dbt and 50% relative humidity. The room air is to be recirculated through a cooling coil except that enough room air is to by pass the cooling coil to satisfy the specific humidity requirements of the supply state. With zero requirements for ventilating air, and assuming that the air which leaves the cooling coil is saturated at the coil surface temperature (apparatus dewpoint), determine:

a. mass flow rate of supply air, pounds of dry air per hour
b. specific humidity of supply air, grains/pound. dry air
c. mass flow rate of air through the cooling coil, pounds of dry air per hour





$$\text{SHR} = \frac{600{,}000}{850{,}000} = 0.705$$

a. $$\text{CFM} = \frac{Q_s}{1.08(t_A - t_c)} = \frac{600{,}000}{1.08(80-65)} = 37{,}000 \text{ CFM}$$

$$4.5 \times 37{,}000 = 166{,}500 \ \frac{\text{lb}}{\text{hr.}}$$

b. From Psychrometric chart at pt. C

Specific Humidity = 66.5 grains per lb. Dry Air

c. From Psychrometric Chart t_{db} at pt. B = 52.2°F

$$\text{CFM} = \frac{Q_s}{1.08(t_A - t_B)} = \frac{600{,}000}{1.08(80-52.2)} = 20{,}000 \text{ CFM}$$

$$4.5 \times 20{,}000 = 90{,}000 \ \frac{\text{lb.}}{\text{hr.}}$$

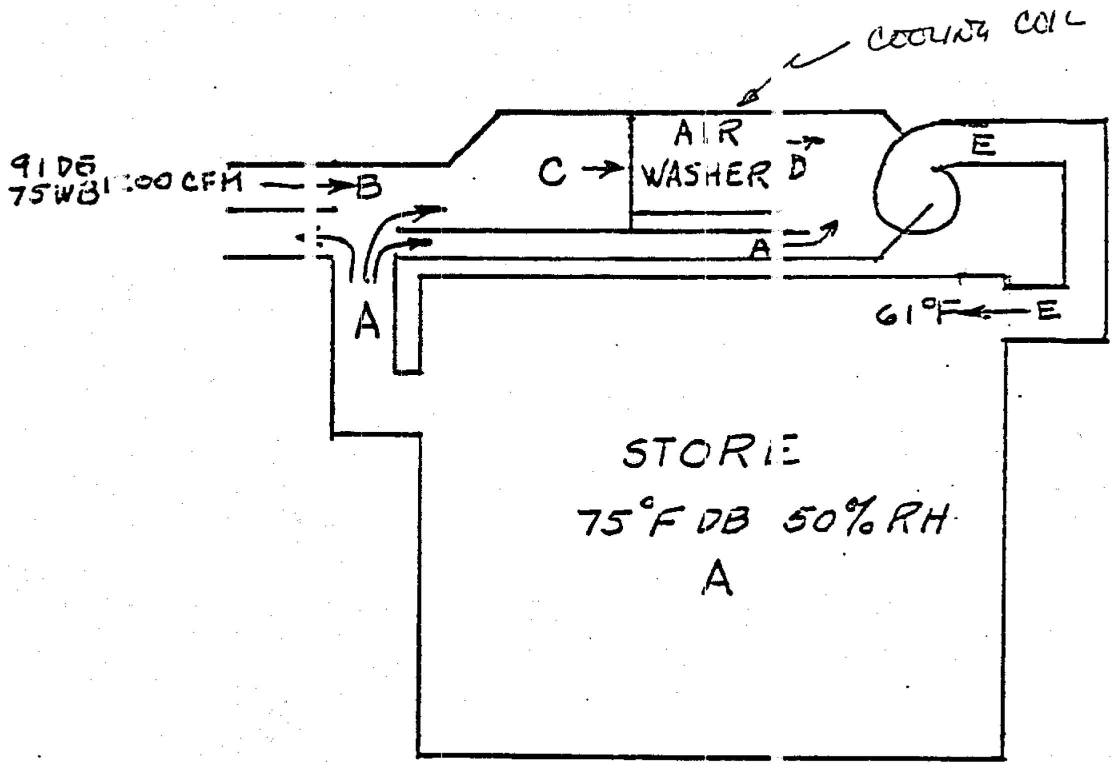
ALL HEAT (Q_s + Q_L) IS REMOVED IN COIL

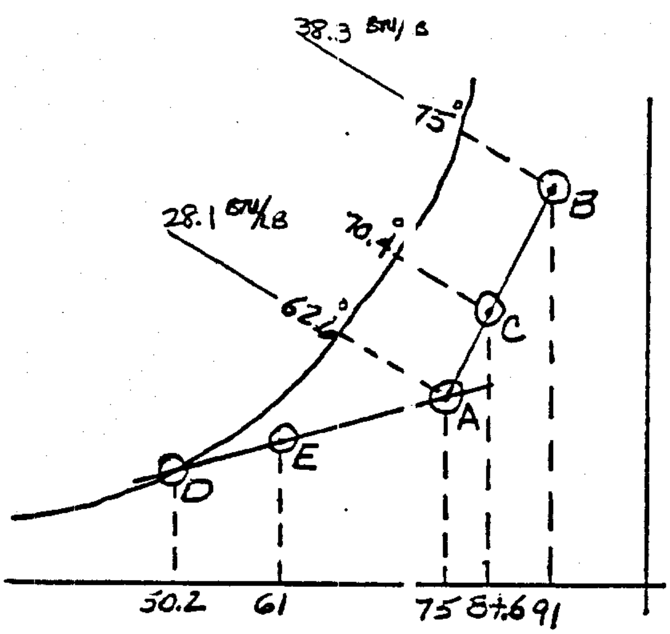
Air by passed around coil would be:

37,000 - 20,000 = 17,000 CFM

A small store has a total heat gain from all sources, except outdoor air, of 67,500 Btu per hr. The latent heat portion of the above is 14,400 Btu per hr. Required indoor air state is to be 75°F dry bulb and 50% relative humidity. The outdoor air state is 91°F dry bulb and 75°F wet bulb. 1200 cfm outdoor air is supplied to the system and mixed with a part of the recirculated air. This mixture then passes through an air washer having an apparatus dew point of 50.5°F. A part of the recirculated air (by-passed air , mixes with the air leaving the washer and the resulting mixture enters the fan which supplies air to the room at a temperature of 61°F. Neglect energy of the fan.

a. Determine the tons of refrigeration required.
b. Determine the volume of by-passed air in cfm.





$67,500 - 14,400 = \quad 53,100 \ \frac{Btu}{HR}$ sensible

$$SHR = \frac{53,100}{67,500} = 0.789$$

Total heat load from outside air (O/A IS LOWERED TO ROOM (NEUTRAL) CONDITIONS) LOAD RESULTS ~~FROM~~ WHEN

$$Q_{TOTAL} = 4.5 \times CFM \times (h_B - h_A)$$
$$= 4.5 \times 1200 \times (38.3 - 28.1)$$
$$= 4.5 \times 1200 \times 10.2$$
$$= 55,100 \ \frac{Btu}{HR.}$$

Total heat load (room and outside air) =

$$\begin{array}{r} 55,100 \\ 67,500 \\ \hline 122,600 \end{array} \ \frac{Btu}{HR.}$$

$$\frac{122,600}{12,000} = 10.2 \text{ tons}$$

Supply Air Required

$$CFM = \frac{Q_S}{1.08(t_A - t_E)} = \frac{53,100}{1.08(75-61)}$$
$$= 3,520 \ CFM$$

Condition "D" air (leaving washer) must remove all heat gain in the store because the by-pass air cannot do any cooling. (same temperature)

$$CFM \ (D, air) = \frac{53,100}{1.08\ (t_A - t_D)}$$
$$= \frac{53,100}{1.08\ (75 - 50.2)}$$
$$= 1,980 \ CFM$$

By-passed air = supply air-washer air

= 3,520 - 1,980

= 1,540 CFM

Another method of determining total heat load (room and outside air) using 1980 CFM for the flow thru the washer, find the condition of the air entering the washer. 1980 CFM - 1200 CFM = 780 CFM of recirculated air.

$\frac{780}{1980} \times 75 = 29.5$

$\frac{1200}{1980} \times 91 = \frac{55.1}{84.6}$ mixture dry bulb entering washer. From psychrometric chart mixture wet bulb = 70.4. Total heat load removed by washer

$$Q_{Total} = 4.5 \times CFM \times (h_c - h_d)$$

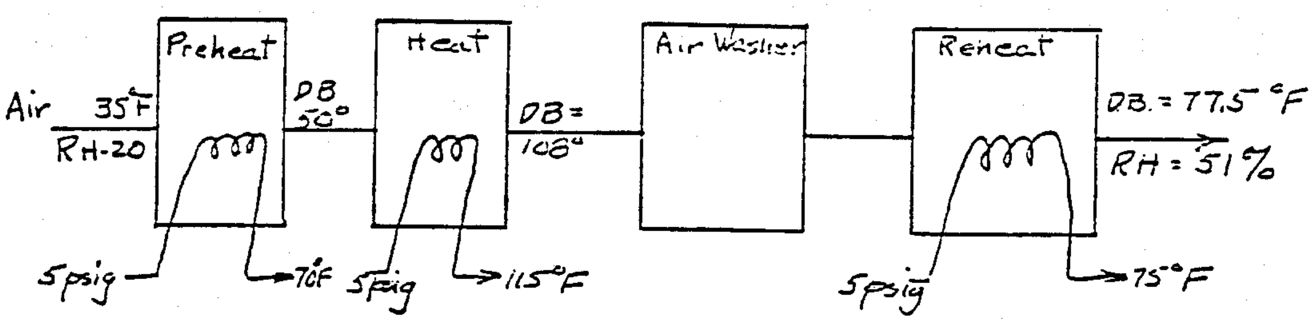
$$= 4.5 \times 1980 \times (34.2 - 20.5)$$

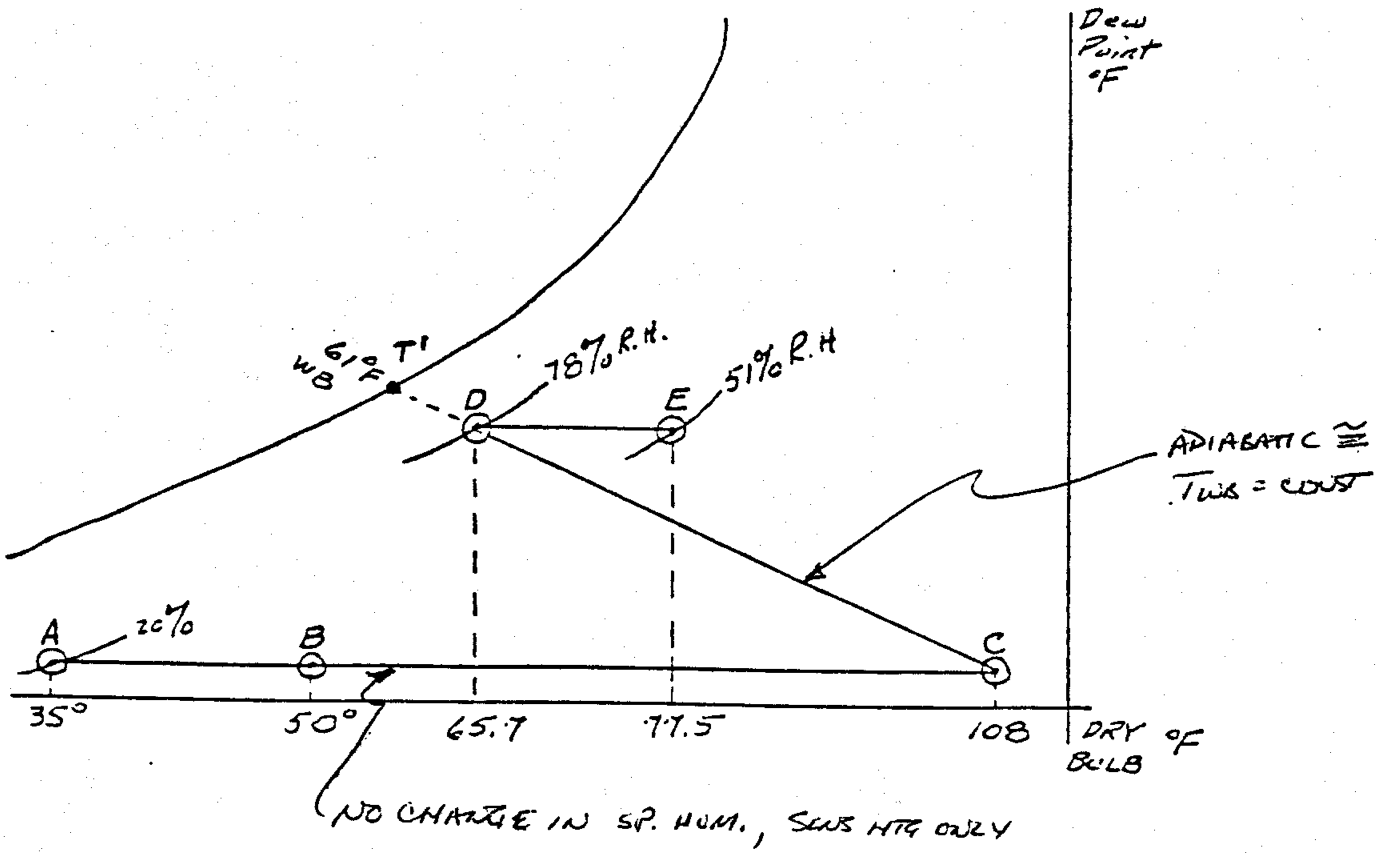
$$= 4.5 \times 1980 \times 13.7$$

$$= 122{,}200 \ \frac{Btu}{hr.}$$

10,000 cfm of air measured at 35°F dry-bulb temperature and a relative humidity of 20% enters a preheat coil and leaves at a dry bulb temperature of 50°F The air then enters a heating coil and leaves at a dry-bulb temperature of 108°F. The air leaves the heating coil and enters an air washer using recirculated water whose saturation efficiency is 90%. The air leaves the air washer and enters a reheat coil. The air leaving the reheat coil is at 77.5°F dry-bulb (and 51% relative humidity) Steam to the heating coils is available 5 psig and saturated. The barometric pressure is 29.92 in Hg. The temperature of the condensate leaving the preheat coil is 70°F; leaving the heating coil is 115°F; and leaving the reheat coil is 75°F.

Determine
(a) the lbm/hr of steam supplied to the preheat coil
(b) The lbm/hr of steam supplied to the heating coil
(c) the lbm/hr of steam supplied to the reheat coil
(d) the gpm of make-up water to the air washer
(e) the dry-bulb temperature and relative humidity of air leaving the air washer





Saturation Efficiency $= \dfrac{t_C - t_D}{t_C - t^1}$

Where

t_C = dry bulb temperature of entering air ^oF

t_D = dry bulb temperature of leaving air ^oF

t^1 = wet bulb temperature of entering air ^oF (SATURATION TEMP)

$$0.90 = \frac{108 - t_D}{108 - 61}$$

$$108 - t_D = 0.90 \times 47$$
$$- t_D = 42.3 - 108$$
$$t_D = 65.7\ ^oF$$

From Psychrometric Chart

$h_A = 9.3 \ \frac{Btu}{lb_{D.A.}}$

$h_B = 12.9$ "

$h_C = 26.8$ "

$h_D = 27.1$ "

$h_E = 30$ "

THERE IS REDUNDANT INFO GIVEN. STATE D IS DETERMINED KNOWING STATE C + SAT. EFF. OF WASHER. POINT E MUST BE ALONG A CONST SP. HUM. LINE ; IF T_{DB} IS SPECIFIED, THEN RH WILL RESULT

NCEE Dec., 1968

Heat added to air by preheat coil

$$Q = 4.5 \text{ CFM } (h_B - h_A)$$
$$= 4.5 \times 10{,}000 \times (12.9 - 9.3)$$
$$= 45{,}000 \times 3.6$$
$$= 162{,}000 \ \frac{\text{Btu}}{\text{hr.}}$$

Note: We are assuming that the 10,000 CFM is at standard air conditions. If it was meant that the CFM should be strictly at condition A then CFM to be used would be

$$10{,}000 \times \frac{13.33}{12.48} = 10{,}700 \text{ CFM}$$ (OR USE MASS FLOW)

$$\dot{m} = \frac{CFM}{v}$$

(a) lb_m/hr of steam supplied the preheat coil

Steam enthalpy entering (5 psig = 20 psia) = 1156.30
Steam enthalpy leaving (70°F) 38.04
= 1118.26 $\frac{\text{Btu}}{\text{lb.}}$

$$\dot{Q} = \dot{m}\Delta h$$
$$\dot{m} = \frac{\dot{Q}}{\Delta h} = \frac{162{,}000}{1118.26} = 144.9 \ \frac{lb_m}{hr.}$$

Heat added to air by heating coil

$$Q = 4.5 \text{ CFM } (h_C - h_B)$$
$$= 4.5 \times 10{,}000 \times (26.8 - 12.9)$$
$$= 45{,}000 \times 13.9$$
$$= 625{,}500 \ \frac{\text{Btu}}{\text{hr}}$$

NOTE: $h_{SAT\,LIQ} \approx T - 32°$

(b) lb_m/hr of steam supplied to the heating coil

steam enthalpy entering = 1156.30
steam enthalpy leaving = 82.93 } FROM STEAM TABLES
AT 115° SAT. LIQ. = 1073.37 $\frac{\text{Btu}}{\text{lb}}$

$$\dot{Q} = \dot{m}\Delta h$$
$$\dot{m} = \frac{\dot{Q}}{\Delta h} = \frac{625{,}500}{1073.37} = 582.7 \ \frac{lb_m}{hr.}$$

Heat added to air by reheat coil

$$Q = 4.5 \text{ CFM } (h_E - h_D)$$
$$= 4.5 \times 10{,}000 \times (30 - 27.1)$$
$$= 45{,}000 \times 2.9$$
$$= 131{,}000 \text{ Btu/hr}$$

(c) lb_m/hr of steam supplied the reheat coil

steam enthalpy entering = 1156.30
steam enthalpy leaving = 43.03 } FROM STEAM TABLES
AT 75° SAT. LIQ. = 1113.27 $\frac{\text{Btu}}{\text{lb.}}$

$Q = \dot{m} \Delta h$

$$\dot{m} = \frac{\dot{Q}}{\Delta h} = \frac{31,000}{113.27} = 117.67 \frac{lb.}{hr.}$$

(d) The gpm of make up water to the air washer

Humidity Ratio @ C = $6 \; grains/lb_{DA}$

Humidity Ratio @ D = $73 \; grains/lb_{DA}$

$$\Delta H.R. = 67 \; grains/lb_{D.A.}$$

Moisture added to air = $\dfrac{4.5 \times CFM \times \Delta H.R.}{7000}$

$$= \frac{4.5 \times 10,000 \times 67}{7000}$$

$$= 430 \; lb/hr$$

$$GPM = \frac{430 \frac{lb}{hr}}{60 \frac{min}{hr} \times 8.33 \frac{lb}{gal}} = 0.859 \frac{gal.}{min.}$$

(e) From Psychrometric Chart

Dry Bulb Temp leaving washer 65.7 °F

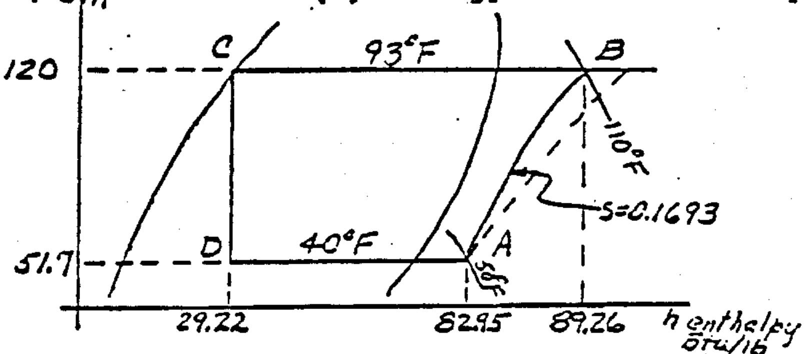
Relative Humidity leaving washer 78%

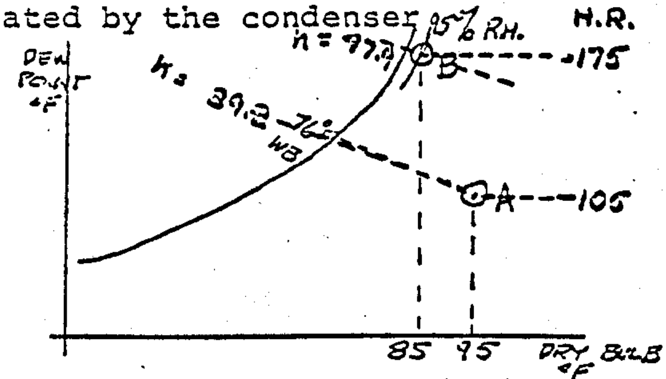
$$4.5 = \frac{FT^3}{MIN} \times 60 \frac{MIN}{HR} \times .075 \frac{LB}{FT^3}$$

CONVERTS CFM → $\frac{LB}{HR}$

An evaporative condenser serving a 25-ton capacity Freon-12 SYSTEM is located on the roof of a building. The outside air is 95°F db, 76°F wb and the air leaves the condenser at 85°F db, 95% relative humidity. The compressor runs with suction conditions of 51.7 psia and 10° of superheat and discharges at 120 psia. The condenser must remove from the refrigerant all the heat added in the evaporator and the heat equivalent of the work done by the compressor. Assuming the actual compressor work to be 30% more than the isentropic work, determine

(a) the condenser load in Btu/min
(b) the required air flow through the condenser in cfm
(c) the gpm of water evaporated by the condenser





Assuming the refrigerant leaves the condenser as a saturated liquid.

Refrigerant flow thru evaporator

$$\dot{m}_{R-12} = \frac{\dot{Q}}{h_A - h_D} = \frac{25 \times 200}{82.95 - 29.22} = \frac{5000}{53.73}$$

$$= 92.9 \ \frac{lb.}{min}$$

Isentropic work of compressor

$$\dot{W} = \dot{m}(h_B - h_A) = 92.9\,(89.26 - 82.95)$$

$$= 92.9 \times 6.31 = 586 \ \frac{Btu}{min}$$

Actual work of compressor

$$W = 1.3 \times \text{isentropic work}$$

$$= 1.3 \times 586$$

$$= 761 \ Btu/min.$$

$$\eta_{comp} = \frac{586}{761} =$$

(a) Condenser Load

Condenser Load = heat added in evap + actual compressor work
= 25 x 200 + 761
= 5,761 $\frac{Btu}{min}$

(b) Required air flow thru condenser

$$CFM = \frac{Q \frac{Btu}{hr}}{4.5(h_B - h_A)} = \frac{5761 \times 60}{4.5(47.9 - 39.2)}$$

$$= 8{,}850 \text{ CFM}$$

(c) gpm of water evaporated by condenser

$$H.R._A = 105 \frac{grains}{lb_{DA}}$$

$$H.R._B = 175 \frac{grains}{lb_{DA}}$$

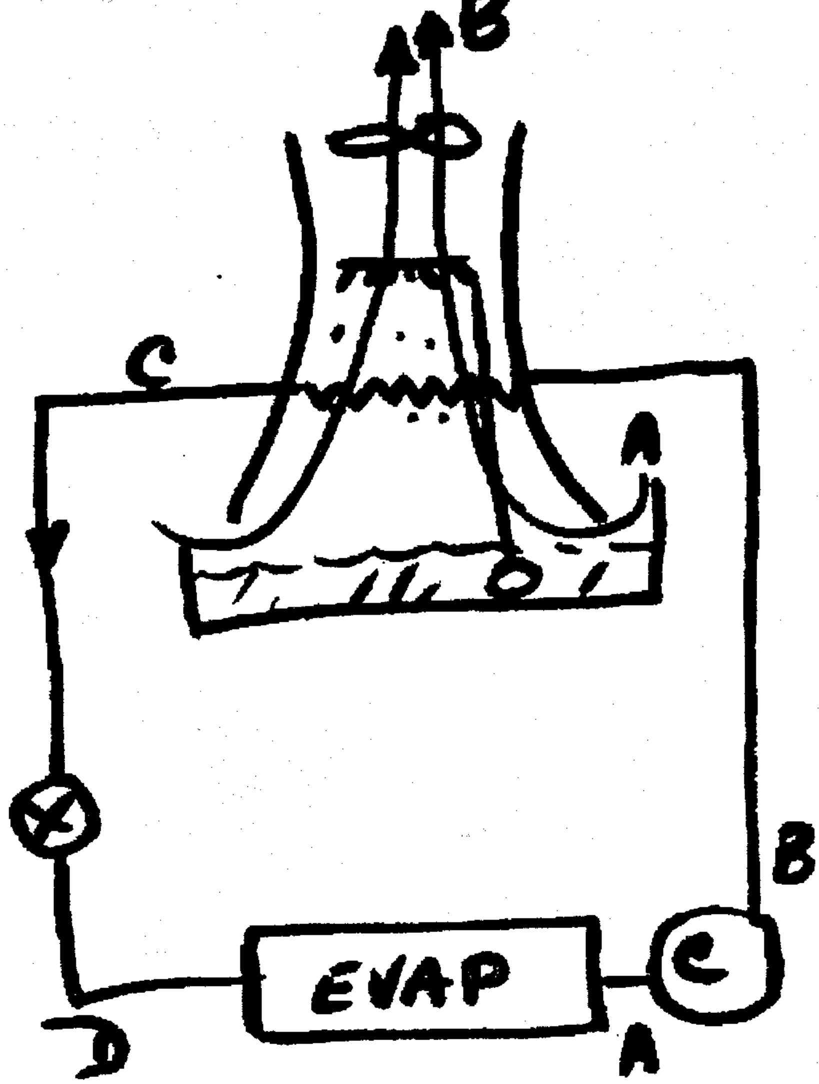
$$\text{Water Flow} = \frac{4.5 \text{ CFM}(H.R._B - H.R._A)}{7000}$$

$$= \frac{4.5 \times 8850 \times 70}{7000}$$

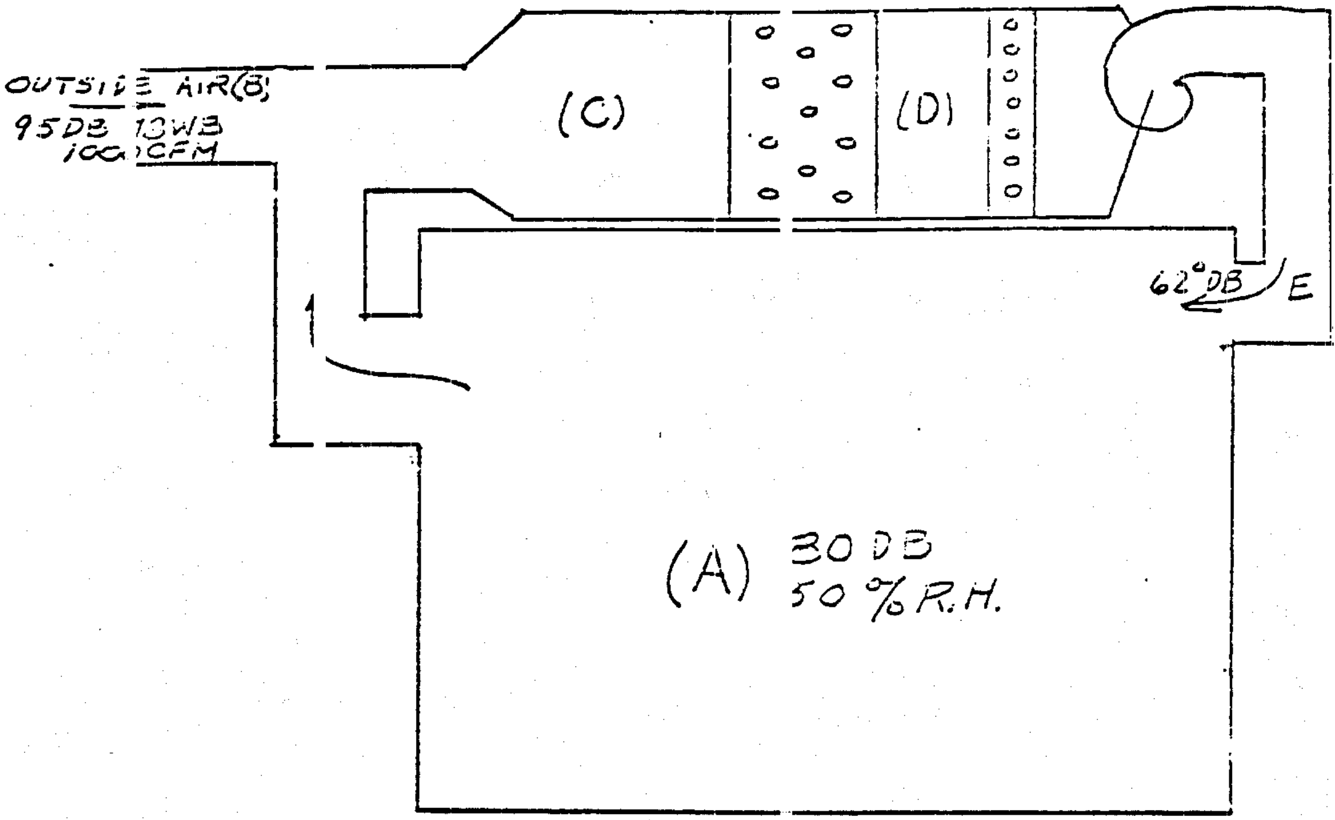
$$= 398 \text{ lb/hr.}$$

$$= \frac{398 \frac{lb}{hr}}{8.33 \frac{lb}{gal}} \times \frac{1}{60 \frac{min}{hr}}$$

$$= 0.797 \frac{gal.}{min.}$$

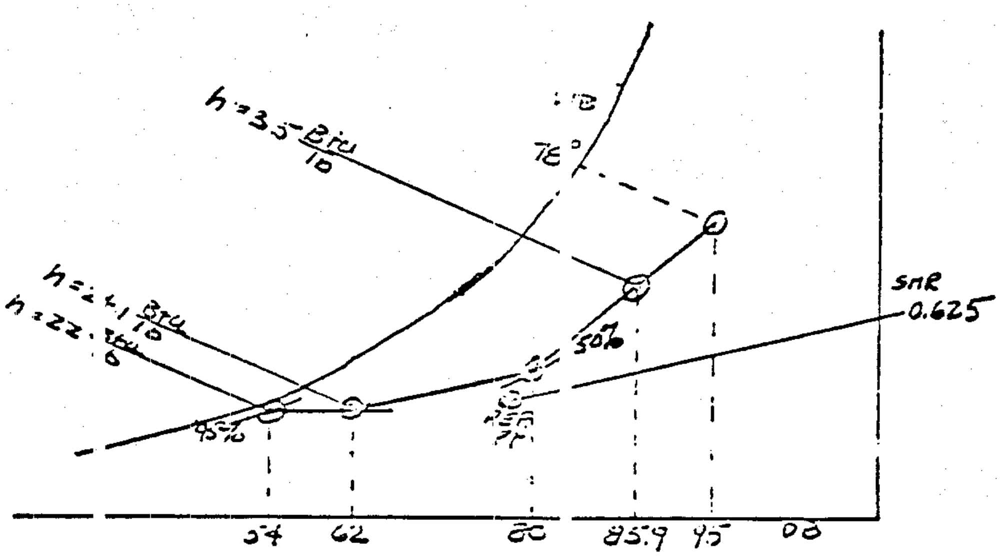

B
C
A
B
EVAP
C
D
A

An industrial building has a sensible heat gain of 50,000 BTUH and a latent heat gain of 30,000 BTUH. 1000 CFM of fresh air is required for ventilation. The space is to be maintained at 80°F and 50% RH, and the supply air is to be at 62° DB. The outside conditions for design are 95° DB and 78° WB. Determine the energy requirements in Btu per hour.



SOLUTION USING RE HEAT

ASSUMED 95 % RH. LEAVING EVAP. COIL.



Sensible Heat Gain 50,000 Btuh
Latent Heat Gain 30,000 Btuh
Total Heat Gain 80,000 Btuh

Sensible Heat Ratio = $\frac{50,000}{80,000} = 0.625$

C.F.M. of Supply Air = $\frac{Q_s}{1.08\,\Delta t} = \frac{50,000}{1.08(80-62)} = \frac{50,000}{1.08 \times 18} = 2,570$ CFM

Finding mixed air conditions (c)

$\frac{1000}{2570} \times 95 = 37.0$

$\frac{1570}{2570} \times 80 = 48.9$

85.9 °F D.B. of air entering coil (c)

Cooling Coil Enthalpy Change
$h_{\text{entering cooling coil}} = 35 \frac{Btu}{lb.}$
$h_{\text{leaving cooling coil}} = 22 \frac{Btu}{lb.}$
$\Delta h = 13 \frac{Btu}{lb.}$

Cooling Coil Capacity $= 4.5 \times CFM \times \Delta h$
$= 4.5 \times 2,570 \times 13$
$= 150,200 \frac{Btu}{HR}$ Heat Removed

Reheat Coil Enthalpy Change
$h_{\text{leaving reheat coil}}$ $24.1 \frac{Btu}{lb}$
$h_{\text{entering reheat coil}}$ $22.0 \frac{Btu}{lb}$
$\Delta h = 2.1 \frac{Btu}{lb}$

Reheat Coil Capacity $= 4.5 \times CFM \times \Delta h$
$= 4.5 \times 2570 \times 2.1$
$= 24,300 \frac{Btu}{HR}$ Heat Added

To Verify Results ——

Outside Air Load:

$h_{\text{outside conditions}} = 41.4 \frac{Btu}{lb}$
$h_{\text{room conditions}} = 31.2 \frac{Btu}{lb.}$
$\Delta h = 10.2 \frac{Btu}{lb}$

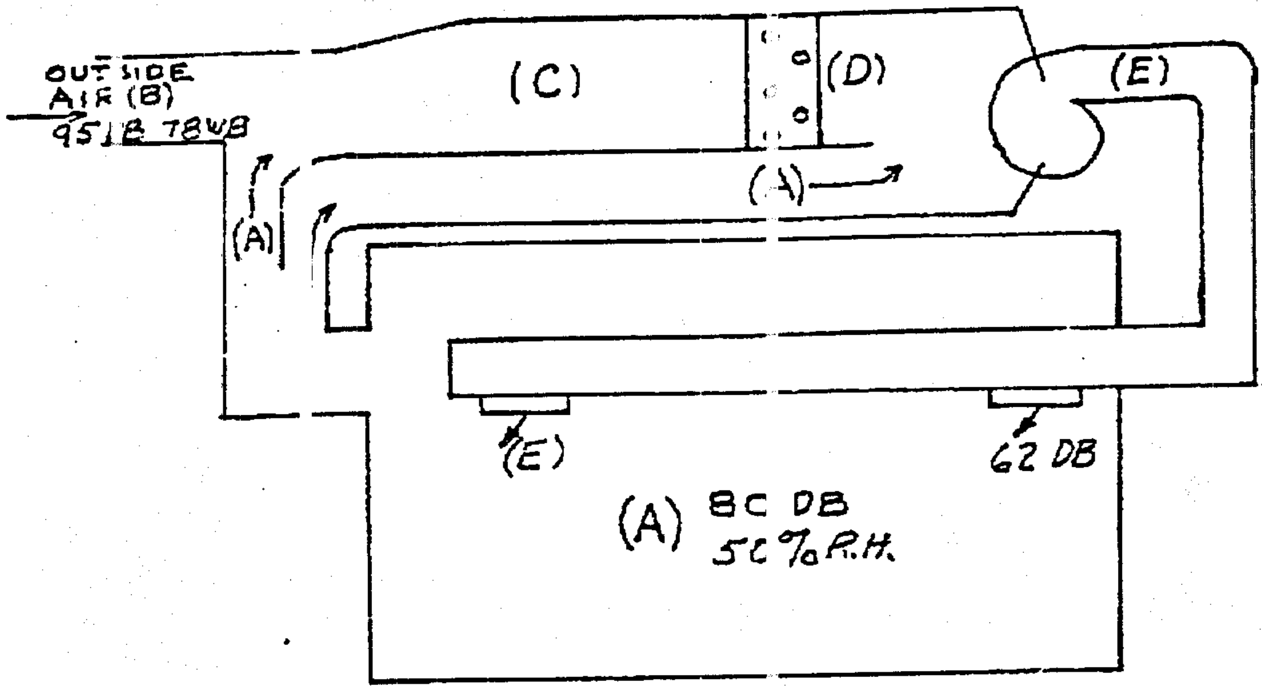
Outside Air Load $= 4.5 \times CFM \times \Delta h$
$= 4.5 \times 1000 \times 10.2$
$= 45,900 \frac{Btu}{HR}$

Total Cooling Load

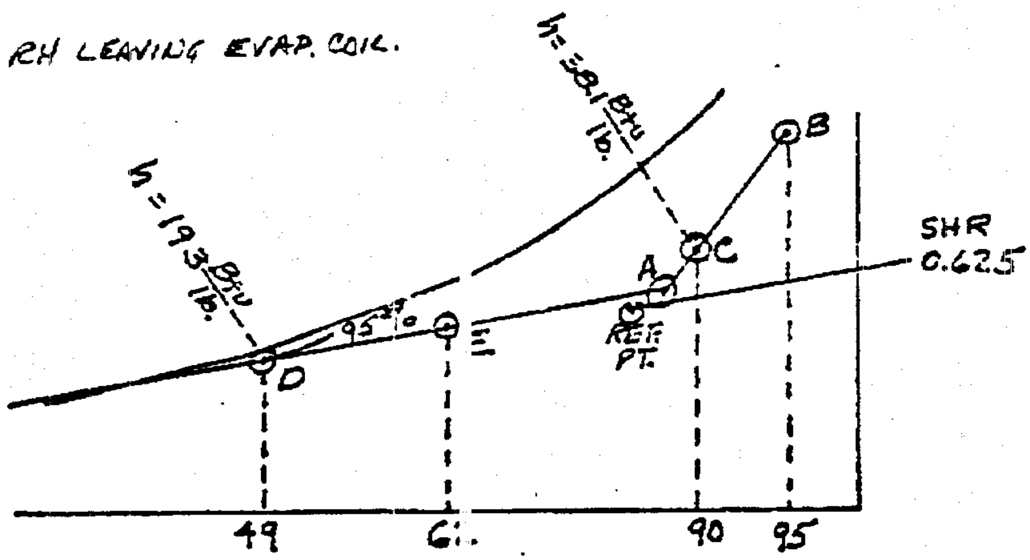
Outside Air	$45,900 \frac{Btu}{HR}$
Room	$80,000 \frac{Btu}{HR}$
Reheat	$24,300 \frac{Btu}{HR}$
	$150,200 \frac{Btu}{HR}$

NCEE APRIL 1970

SOLUTION WITHOUT REHEAT (USING BYPASS AIR)



ASSUMED 95 % RH LEAVING EVAP. COIL.



NOTE:

ALL O/A GOES THRU COIL

IF SHR IS VERY LOW, THIS MAY NOT BE POSSIBLE

NCEE APRIL 1970

SOLUTION WITHOUT REHEAT

CFM THRU COIL

$$CFM = \frac{Q_s}{1.08\Delta t} = \frac{50,000}{1.08(80 - 49)}$$

(TA - TD)

$$= 1493 \text{ CFM}$$

FINDING MIXED AIR CONDITION

(C)

$$\frac{1000}{1493} \times 95 = 63.6$$

$$\frac{493}{1493} \times 80 = \frac{26.4}{90.0} \text{ D.B.}$$

COOLING COIL ENTHALPY CHANGE

$h_{\text{entering coil}}$ (C) = 38.1

$h_{\text{leaving coil}}$ (D) = 19.3

Δh = 18.8 Btu/lb.

COILING COIL CAPACITY = 4.5 x CFM x Δh

= 4.5 x 1493 x 18.8

= 126,307 Btu/hr.

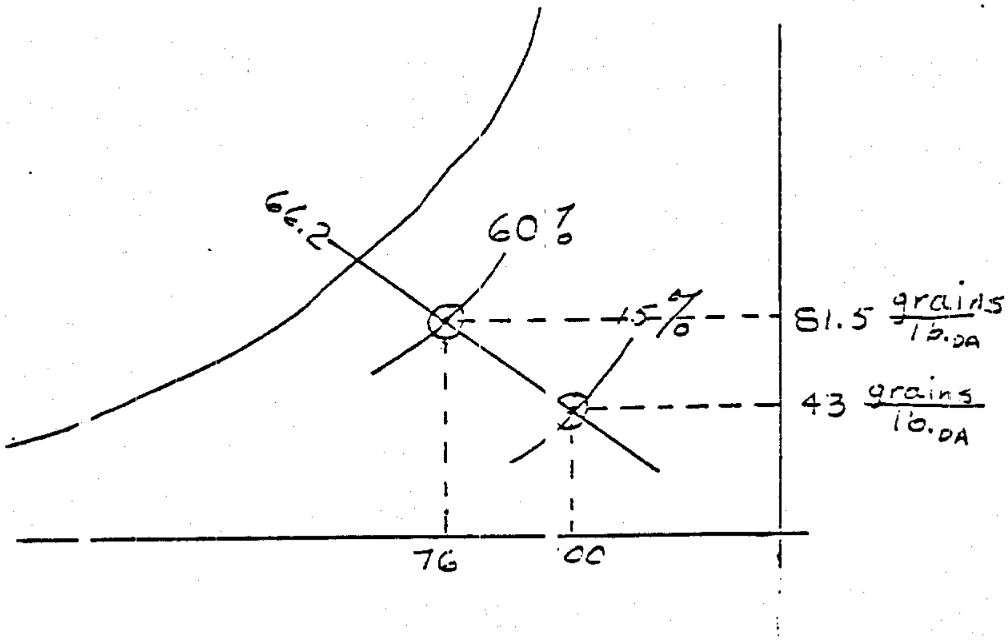
To Verify Results

Outside Air	Q_T	= 45,900 (previous calculation)
Room	Q_T	= 80,000
Total	Q_T	=125,900

SAME AS BEFORE WITHOUT THE R/H

NCEE DEC. 1967

The outside air temperature is 100 degrees F at 15% R.H. It is desired to cool 1500 CFM to the lowest possible temperature by evaporative cooling with the R.H. increasing to 60%. How many pounds of water per minute is required and what is the final temperature?



The solution is based on 1500 CFM of standard air

(a) Pounds of water per minute

$$\dot{m}_{H_2O} = \frac{4.5 \times CFM\,(HR_2 - HR_1)}{7000}$$

$$= \frac{4.5 \times 1500\,(81.5 - 43)}{7000}$$

$$= 37.125\ \frac{lb}{hr} \text{ or } 0.618\ \frac{lb.}{min.}$$

(b) Final temp

76°F (from chart)

LOAD CALCULATION PROBLEM

A 10,000 FT^2 CHURCH SEATS 1000 PEOPLE AND IS TO COOLED TO INDOOR DESIGN CONDITIONS OF 76°F DB AND 64°WB. THE TRANSMISSION LOAD INCLUDING SOLAR LOADING IS 400,000 BTU/HR. LIGHTING AVERAGES 3 WATTS/FT^2 AND IS A MIXTURE OF INCANDESCENT AND FLOORESCENT.

OUTDOOR AIR DESIGN CONDITIONS ARE 95°F DB AND 76°F WB. FRESH AIR IS TO BE PROVIDED AT NORMAL "THEATER" VENTILATION RATES.

CALCULATE :

A. SENSIBLE, LATENT AND TOTAL SPACE COOLING LOADS

B. TOTAL LOAD ON THE COOLING COIL AND COIL SHR

SOLUTION

A. SPACE COOLING LOAD

1. SENSIBLE

TRANSMISSION (GIVEN) 400,000 BTU/HR

LIGHTING - ASSUME ½ INCANDESCENT AND THAT BALLAST IS IN CONDITIONED SPACE

$$LIGHTS = 3 \times \frac{10000}{2} \times 3.413 + 3 \times \frac{10000}{2} \times 3.413 \times 1.2$$

$$= 112600$$

PEOPLE - ASSUME AVERAGE GROUPING AT REST

$$Q'_S = 1000 \times 230 = 230\,000$$

SENSIBLE = 400 000 + 112 600 + 230 000 = 1372600 BTU/HR

2. LATENT

PEOPLE $Q_L = 1000 \times 120 = 120\,000$

3. TOTAL = 1 372 600 + 120 000 = 1492600 BTU/HR = 124.4 TONS

$$SHR = \frac{1372600}{1492600} = .92$$

B. TOTAL LOAD ON COIL

ADDITIONAL LOAD IS DUE TO REQUIRED VENTILATION AIR

CFM = 1000 x 15 CFM/PERSON = 15 000

$$Q_{SENS} = CFM \times 1.08\,(T_{OA} - T_{DES})$$

$$= 15000 \times 1.08\,(95-76)$$

$$= 307\,800 \text{ BTU/HR}$$

TO FIND OUTSIDE AIR LATENT LOAD, THE SPECIFIC HUMIDITIES MUST BE FOUND FROM A PSYCH. CHART

@ 76/64 $\quad W = 70$ gr/lb

@ 95/76 $\quad W = 105$ gr/lb

$$Q_{LAT} = CFM \times .68 \times (W_{OA} - W_{RCS})$$

$$= 15000 \times .68\,(105 - 70) = 357000 \text{ BTU/HR}$$

TOTAL COIL LOAD

$$Q_{SENS} = 1372600 + 307800 = 1680400$$

$$Q_{LAT} = 120000 + 357000 = 477000$$

$$Q_{TOT} = 2157400 \text{ BTU/HR}$$

179.8 TONS

$$SHR = \frac{1680400}{2157400} = .78$$

7/0?S

PE Review- Air Conditioning

A practical method to establish a Sensible Heat Ratio (SHR) line on a psychometric chart.

Background: The SHR line indicates the loci of points along which conditioned air <u>must</u> be delivered to a conditioned space (in proper quantity) to match the design load and hence produce the desired room conditions.

SHR= Qsens/Q total = Qsens/(Qsens+Qlatent)

Using simplified approximate equations:

$$\text{Qsens} = 1.08 * \text{CFM} * \Delta\text{Tdb}$$

$$\text{Qlatent} = .68 * \text{CFM} * \Delta w(grains/lb)$$
$$= 4760 * \text{CFM} * \Delta w(lb/lb)$$

$$\text{SHR} = \frac{1.08 * \text{CFM} * \Delta\text{Tdb}}{1.08 * \text{CFM} * \Delta\text{Tdb} + .68 * \text{CFM} * \Delta w}$$

Rearranging:

$$\Delta w = \frac{(1 - \text{SHR}) * 1.08 * \Delta\text{Tdb}}{.68 * \text{SHR}} \quad \text{(gr/lb)}$$

$$= \frac{(1 - \text{SHR}) * 1.08 * \Delta\text{Tdb}}{4760 * \text{SHR}} \quad \text{(lb/lb)}$$

Procedure:

1. Plot point A, the known design conditions.

2. For a convenient increment of ΔTdb (say 10 or 20 deg), calculate Δw with the known SHR.

3. Calculate conditions for point B:

$$Tdb = T_{Adb} + \Delta Tdb$$
$$w = w_A + \Delta w$$

4. Plot point B

5. Connect points A and B to establish the SHR line.

Example:

Design conditions: Tdb=78 degF, RH = 50%, SHR=.8

From psych chart : w = 71.5 gr/lb (.01022 lb/lb)

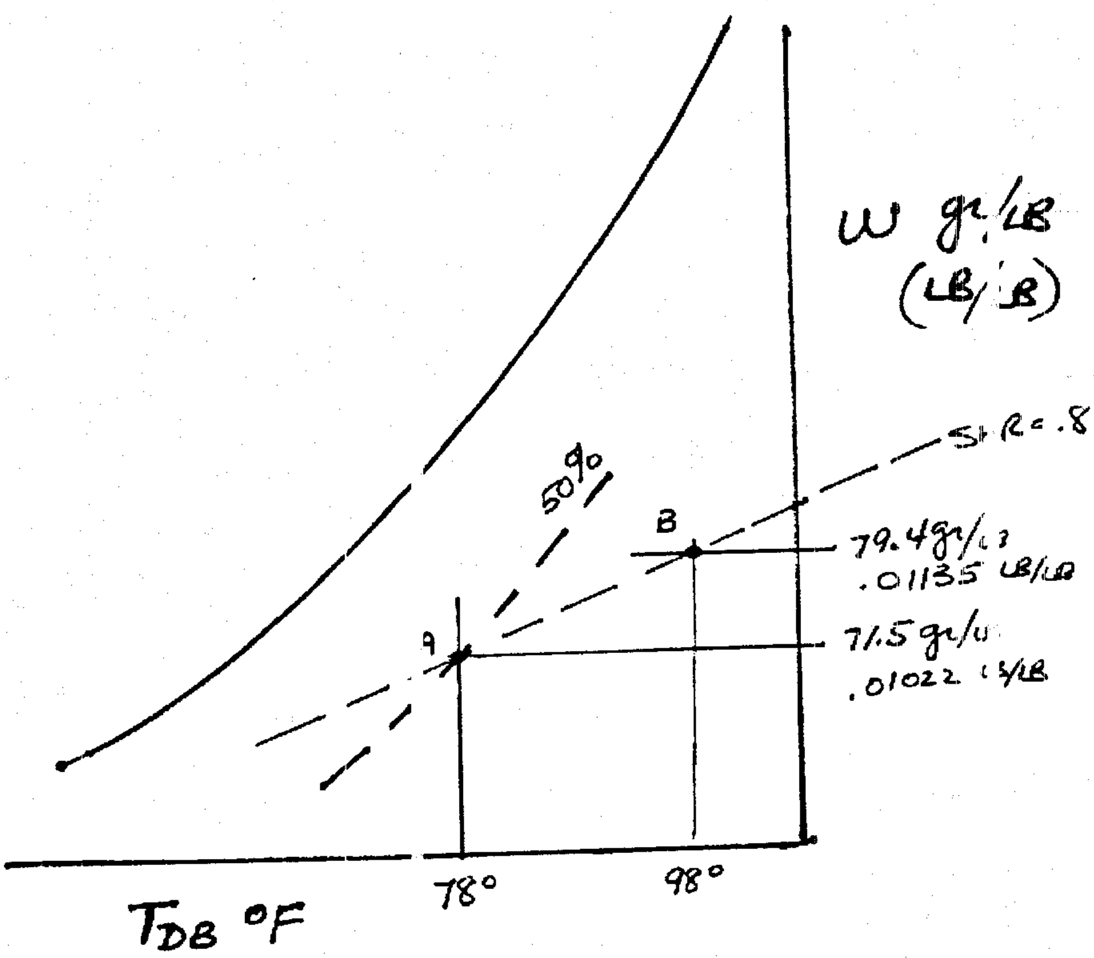
For:

$$\Delta T = 20 \text{ deg}$$
$$\Delta w = \frac{(1-.8)*1.08*20}{.68*.8} = 7.9 gr / lb \quad (.001134 \text{ lb/lb})$$

Therefor:

$$T\,db = 78 + 20 = 98 degF$$
$$w = 71.5 + 7.9 = 79.4 gr / lb \quad (.01135 \text{ lb/lb})$$

Since the selected design point is the "bulls eye" on the Trane Psych chart in this case we can check to see where the SHR line falls on the SHR scale.

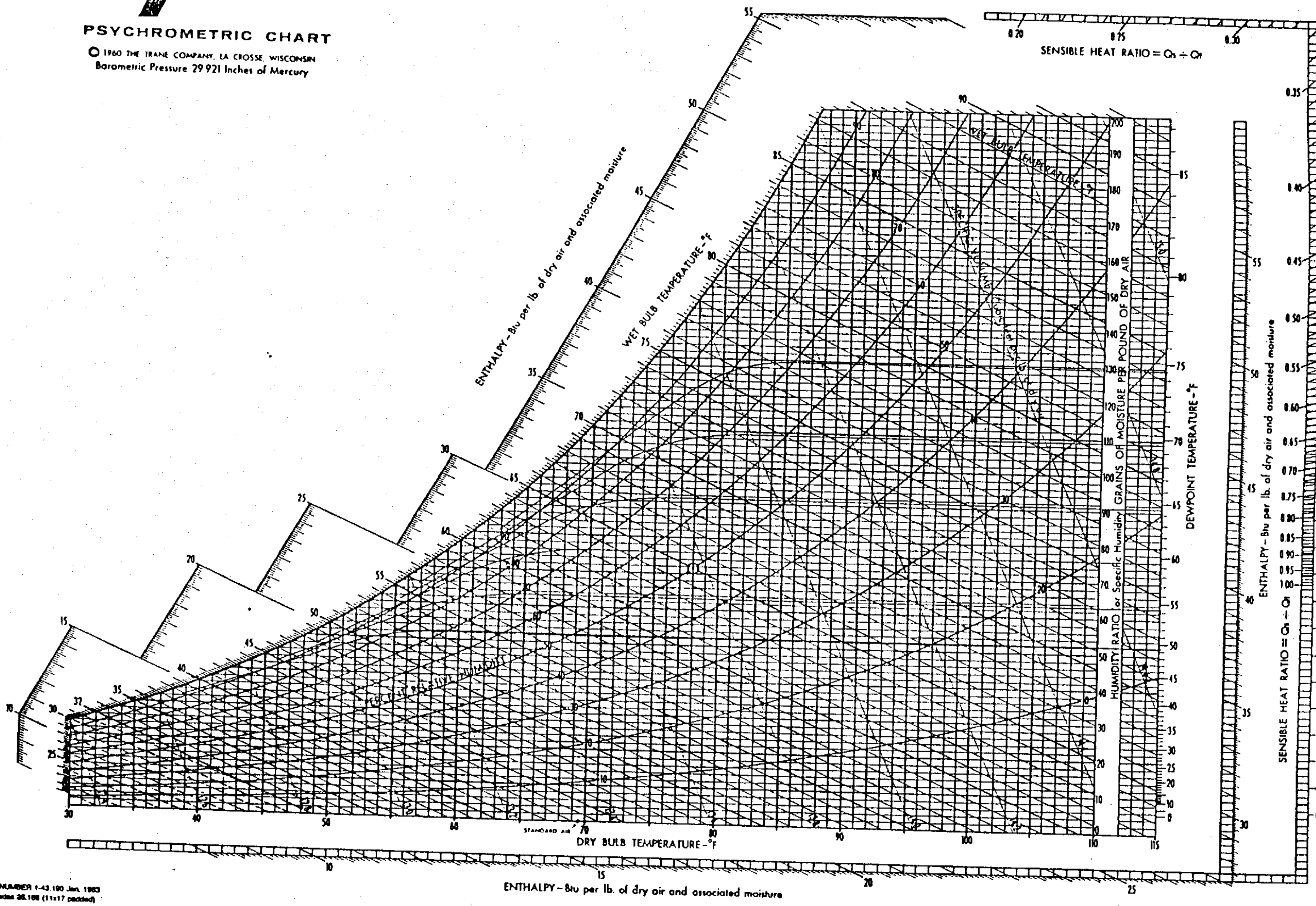


ω gr/LB
(LB/LB)
SHR = .8
50%
B
79.4 gr/LB
.01135 LB/LB
A
71.5 gr/LB
.01022 LB/LB
78°
98°
TDB °F

TRANE®

PSYCHROMETRIC CHART

Barometric Pressure 29.921 Inches of Mercury



FORM NUMBER 1-43.190 Jan. 1963
Supersedes 26.188 (11x17 padded)

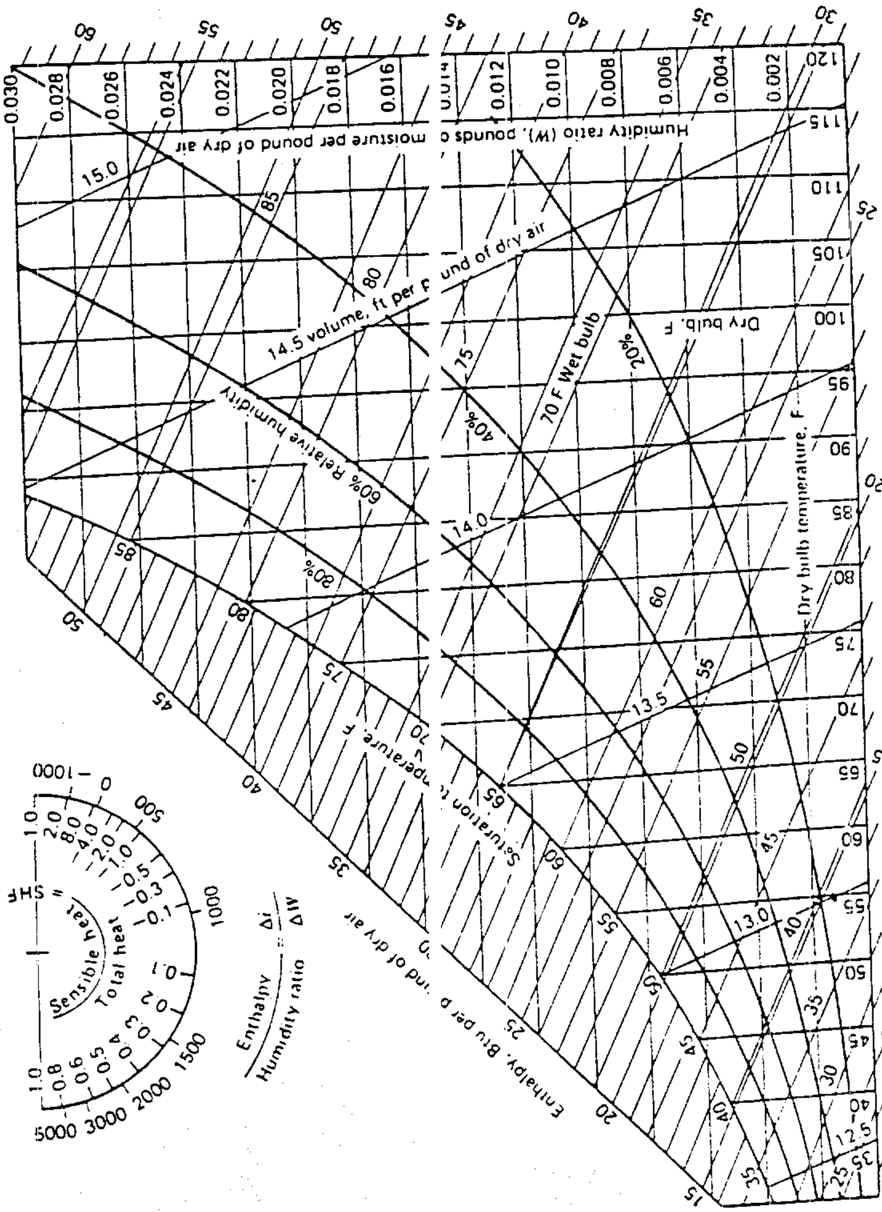
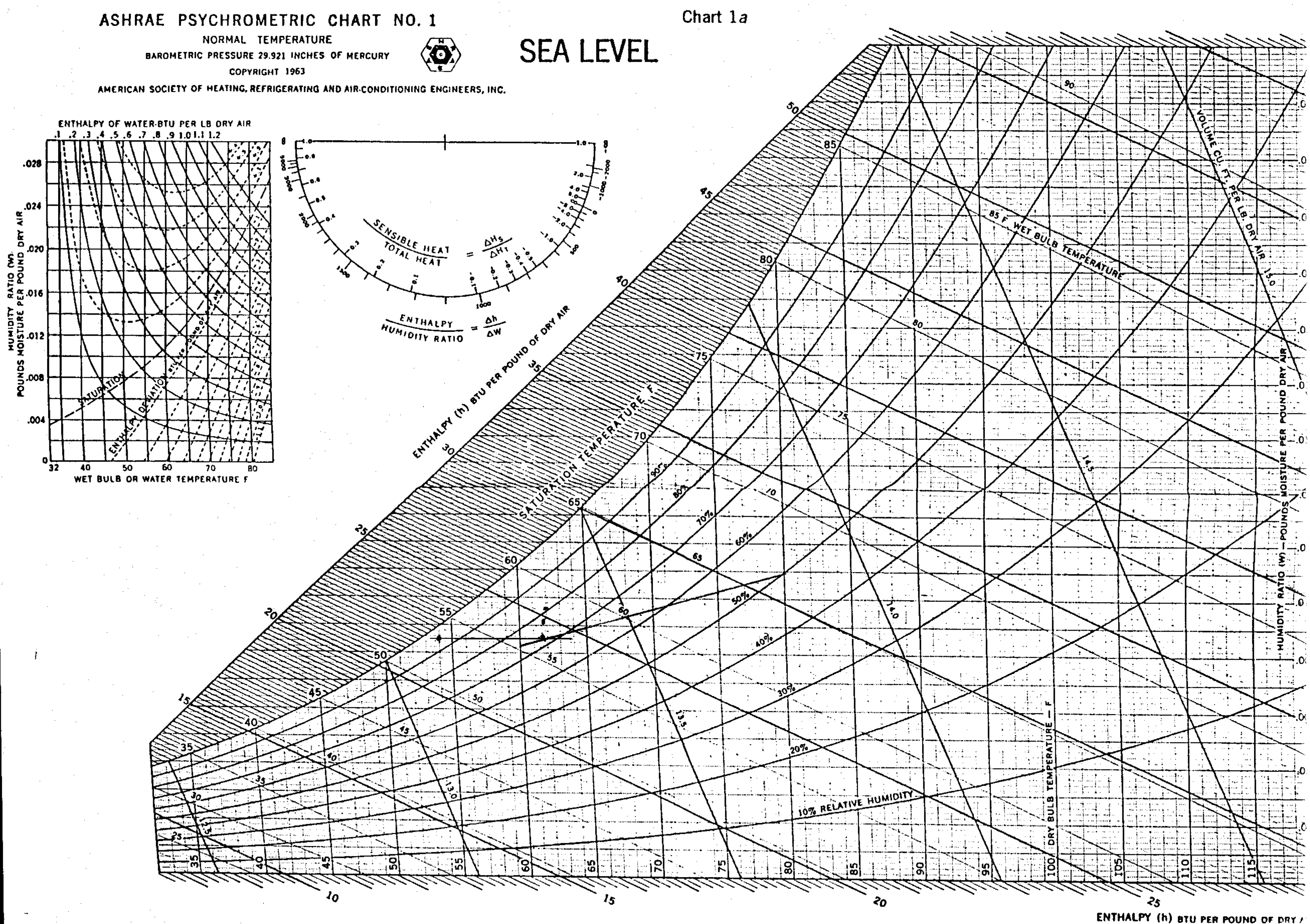


Figure 3-6 Abridgement of ASHRAE psychrometric Chart 1a. (Reprinted by permission from ASHRAE Handbook, Fundamentals Volume, 1989.)

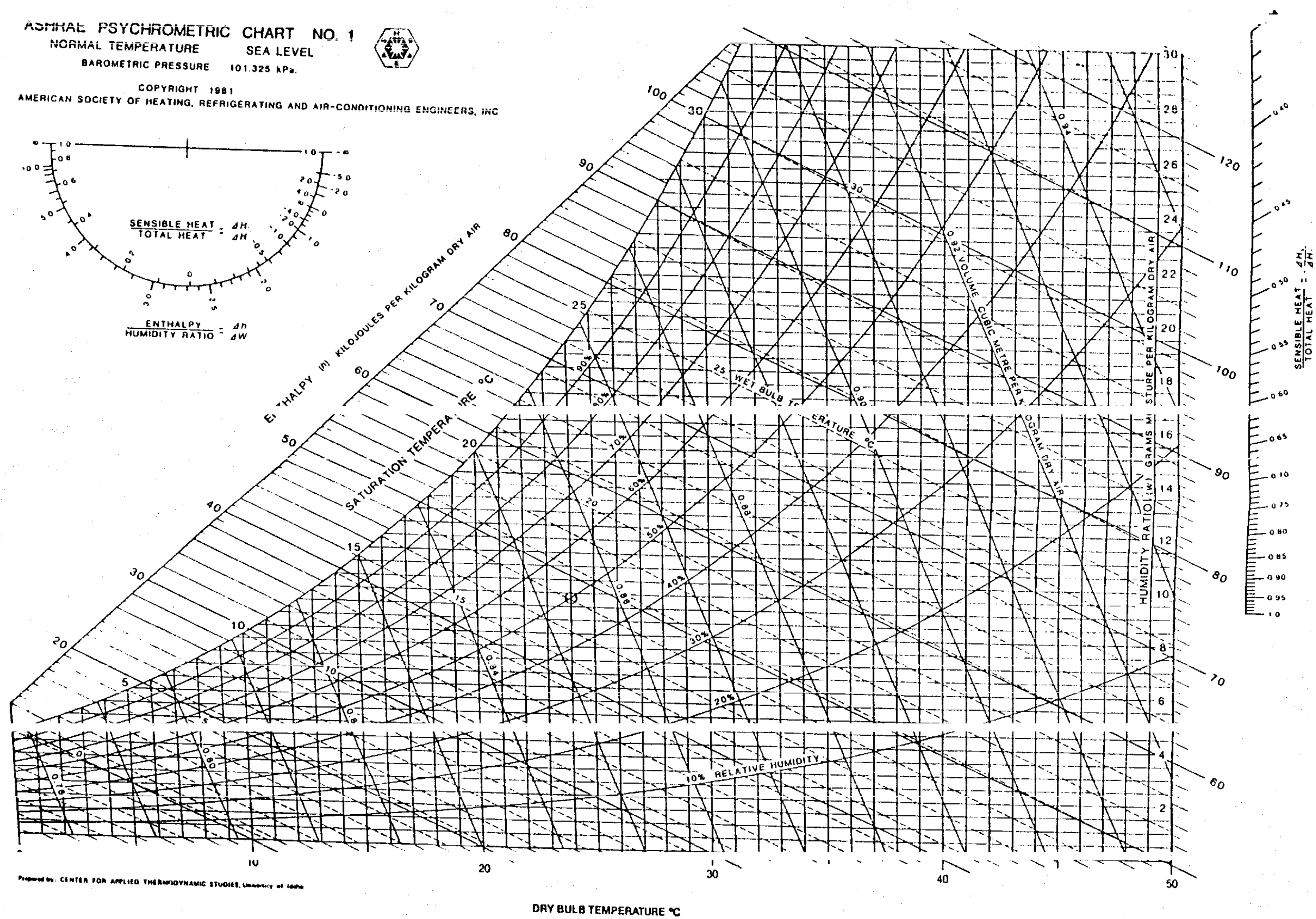
3-5

Heati:

Chart 1a



ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
BAROMETRIC PRESSURE 29.921 INCHES OF MERCURY
COPYRIGHT 1963
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC.
SEA LEVEL
ENTHALPY OF WATER-BTU PER LB DRY AIR
HUMIDITY RATIO (W) POUNDS MOISTURE PER POUND DRY AIR
WET BULB OR WATER TEMPERATURE F
SATURATION
ENTHALPY DEVIATION
SENSIBLE HEAT / TOTAL HEAT = ΔHs / ΔHt
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY (h) BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE F
85 F WET BULB TEMPERATURE
VOLUME CU. FT. PER LB. DRY AIR
10% RELATIVE HUMIDITY
DRY BULB TEMPERATURE — F
HUMIDITY RATIO (W) — POUNDS MOISTURE PER POUND DRY AIR
ENTHALPY (h) BTU PER POUND OF DRY /



ASHRAE PSYCHROMETRIC CHART NO. 1
NORMAL TEMPERATURE
SEA LEVEL
BAROMETRIC PRESSURE 101.325 kPa.
COPYRIGHT 1981
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS, INC
SENSIBLE HEAT / TOTAL HEAT = ΔH / ΔH
ENTHALPY / HUMIDITY RATIO = Δh / ΔW
ENTHALPY (h) KILOJOULES PER KILOGRAM DRY AIR
SATURATION TEMPERATURE °C
WET BULB TEMPERATURE °C
0.92 VOLUME CUBIC METRE PER KILOGRAM DRY AIR
10% RELATIVE HUMIDITY
HUMIDITY RATIO (W) GRAMS MOISTURE PER KILOGRAM DRY AIR
DRY BULB TEMPERATURE °C
SENSIBLE HEAT / TOTAL HEAT = ΔH / ΔH
Prepared by: CENTER FOR APPLIED THERMODYNAMIC STUDIES, University of Idaho

Chart 1b